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THE THERMO-DYNAMIC PRINCIPLES OF ENGINE DESIGN

L. M. Hobbs

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PREFACE.

THIS book has been written with the object of placing in a concise manner, before engineers and students, the thermodynamic principles underlying the design of various forms of heat engines. With this object in view, no detailed description of the mechanisms of the various types referred to has been attempted except in so far as has been necessary, to explain the theoretical principles involved. To render the book of more use to students, a number of examples have been added as an Appendix.

The calculation of heat quantities by means of the *temperature-entropy* diagram has been purposely avoided throughout, as it has been thought better, for the sake of uniformity, to treat every type from the pressure-volume point of view only.

I desire to take this opportunity of acknowledging my indebtedness to Professor S. Dunkerley, now of Manchester University, for much information and kindly help—both formerly when his pupil, and also more recently while his assistant at the Royal Naval College, Greenwich.

L. M. HOBBS.

GREENWICH, *December* 1906.

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THE
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CHAPTER I.

LAWS AND PRINCIPLES OF THERMO-DYNAMICS.

§ 1. **First and Second Laws of Thermo-dynamics.**— Thermo-dynamics includes the consideration of the relations between the quantity of heat supplied to given substances and the amount of mechanical work that can be obtained from them in return.

The present chapter will be devoted to considering some of the principal laws that govern the changes of heat into mechanic energy, and *vice versâ*; a full appreciation of these laws being required for the proper study of the subject.

There are two principal laws of thermo-dynamics justified by observation and experience; they may be stated as follows:—

First Law of Thermo-dynamics.—Heat and mechanical energy are mutually convertible. In other words, whenever work is performed by the agency of heat, an amount of heat disappears equivalent to the work done; and conversely, whenever mechanical work is employed in generating heat, the heat generated is equivalent to the work performed.

Second Law of Thermo-dynamics.—It is impossible for a self-acting machine, unaided by external energy, to convey heat from one body to another at a higher temperature. This law is sometimes stated in a rather different form; but the above statement,

due to Clausius, contains the principle involved, and will be sufficient for the purposes of this book.

§ 2. **Consideration of the First Law.**—It is found experimentally that heat produces by its disappearance—and requires for its production—mechanical energy equivalent to 774* foot-lbs. of work for each thermal unit so lost or produced, the latter being defined as the amount of heat necessary to raise 1 lb. of water at about 39° F. through one degree on the Fahrenheit scale. The First Law states, therefore, that if a body receives an amount of heat from any source equal to H thermal units, the exact mechanical work equivalent to this is $J \times H$ foot-lbs. where $J = 774$, and is known as *Joule's equivalent*, or, more shortly, as "the Joule." It states no more than that $J \times H$ is the amount of work that can be got out of the substance if there are no losses.

Now, in doing work, a substance undergoes some cycle of energy changes, and if the cycle be a complete one, the substance eventually returns to its initial state. Its stock of energy is then precisely the same as at first, so that whatever heat has been received, none of it remains in the body. But work will have been done at the expense of the heat received, and the First Law merely states that the work done is exactly equal to the heat received if the efficiency be unity.

§ 3. **Efficiency.**—By *efficiency* we therefore mean the ratio of the actual work done to the heat supplied.

Thus in nearly every form of thermal engine mechanical work is obtained by means of the force of the expansion of an elastic fluid, acting usually on a piston travelling in a cylinder. Heat is originally supplied to the elastic fluid to cause the expansion, and of this heat part is lost by conduction to the containing walls, and another part is carried away when the fluid escapes to the air or condenser at the end of the stroke of the piston, but a third part has disappeared altogether as heat. This third portion is the exact equivalent of the work done by the elastic fluid in driving

* Joule has given this value as 772 foot-lbs., and more recently it has been shown to be 778 foot-lbs.; but the above value is given by Regnault, and since the steam tables in general use were compiled by him and involve this constant, it is evident the value 774 is the one to take, and it will be used, therefore, throughout this book.

the piston. It is the ratio of this third portion (which is converted into actual useful work) to the heat originally supplied to the fluid which is a measure of the efficiency of an engine.

We must bring the substance back to its initial state to discuss the question of its efficiency; for though we can go from one state to another and find the heat received and work done during the change, we cannot talk of the efficiency, because part of the heat received has gone to increase the stock of internal energy of the substance, and is, therefore, still available.

In practice, the working fluids used in engines always pass through a closed heat cycle, and eventually come back to their original condition ready to repeat the cycle again; as, for instance, the steam in a steam-engine cylinder. It starts as cold water on entering the boiler, is there converted into steam, is used in the engine as such, and finally becomes condensed to water again in the condenser ready to be pumped back into the boiler once more.

The actual thermal efficiency of engines is always small—not greater than 20 per cent. for a steam-engine, 30 to 35 per cent. for a gas or oil engine, and 40 per cent. for an air-engine, though the last is never realised for mechanical reasons.

What, then, has become of the remainder of the heat originally supplied to the working fluid? It is not in the substance, for that is assumed to be again in its initial state. It must have been imparted to outside objects, as the cylinder, condenser, etc., for example. We thus have to measure three things in any cycle of operations, viz.: heat received, heat converted into useful work (*i.e* work done), and heat rejected or lost.

We can represent these very simply graphically.

§ 4. Graphical Representation of Heat Quantities.—Suppose we have a substance initially in a state A so that its pressure is represented by AE and its volume by OE, and suppose heat to be supplied to the substance so that it expands in some manner till it reaches the state B. Then the work done during the operation AB by the expanding substance is obviously the area ABFE in fig. 1. Now draw through the points A and B adiabatic* curves, which will

* An "adiabatic curve" of expansion or compression is the curve connecting

eventually touch the axis of volumes at infinity. Let these be AM and BN. Draw CDG through any point C on the curve BN, vertically to OG. Then we can look on ABCD as an indicator card, and the substance will come back to its initial state A, after having performed the operations AB, BC, CD, and DA. Now, during AB heat has been received by supposition. During BC no heat is received since it is an adiabatic curve, and similarly during DA, nor is any heat rejected. Hence since the substance returns to its initial state A, heat can only have been rejected during CD. But external work has been done, measured, as has

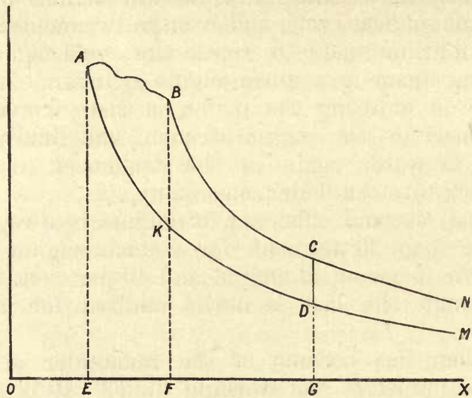

FIG. 1.

just been seen by the area ABCD. Hence the heat rejected during CD must be the difference between the heat received during AB and the work measured by ABCD. If we take the point C nearer N, more work will be done (since the area ABCD will increase) for the same heat received; and finally, if the point C be at an infinite distance along the curve BN, the points C and D will coincide and no heat at all will be rejected. The area ABNM when N and M are at infinity will represent, therefore, the whole of the heat received during AB. We have thus obtained an area ABFE to represent the work done during any operation as AB,

the pressures and volumes of a gas when a constant quantity of it is being expanded or compressed in a non-conducting cylinder without receiving heat from, or imparting heat to, any external objects.

and an area ABNM to represent the heat received during the same operation. Therefore:—

$$\begin{aligned} \text{Heat received} - \text{Work done during any such operation} \\ &= \text{area MABN} - \text{area ABFE} \\ &= \text{area MKBN} - \text{area AKFE} \\ &= \text{area NBFX} - \text{area MAEX.} \end{aligned}$$

This difference is independent of the path AB, so that when a substance passes from a state A to another state B, the difference between the total heat received and the external work done depends purely and simply on its initial and final states.

Now the area MAEX represents the amount of work that could be got out of the substance at state A if it were expanded adiabatically to zero pressure, and is called the *intrinsic energy* of the substance at state A. So also the area NBFX represents the intrinsic energy of the substance at the state B.

Therefore we have:—

$$\text{Heat received} - \text{Work done} = \text{Increase or decrease of intrinsic energy.}$$

If the substance pass through a complete cycle of operations, coming back to the state A, as in fig. 1A, heat is rejected during ADB measured by the area MADBN; and work is done, represented by the shaded area ACBD. Since the substance returns to its original state A, it has the same intrinsic energy finally as before, and has therefore merely been an agent for converting heat into useful work. The efficiency (η) is then given by

$$\eta = \frac{\text{area ABCD}}{\text{area MACBN}},$$

and the heat rejected, represented by MADBN, is lost.

Why has this heat to be rejected? Even if our mechanical appliances will not permit of our converting more than a certain proportion of the heat received into work, why cannot we make this heat rejected flow back into the substance and so save it? For the reason that heat will not of its own accord flow from a lower to a higher temperature, which is the Second Law of thermo-dynamics. This law, therefore, imposes a limit on the application of the First Law.

Consider the useful work area ABCD in fig. 1A. Suppose we draw an isothermal* curve corresponding to the lowest temperature we have available. Let TT′ be this isothermal curve. Then we can increase our indicator card ABCD by extending it to ABT′T; but this is the maximum amount of work that we can get from the heat received during AB, for even if we form an artificially lower temperature and so get an isothermal SS′, and an increase of useful work measured by TT′S′S, we should have first to produce this artificial lower temperature, and in producing it would have to do more work than we should get back from our engine by

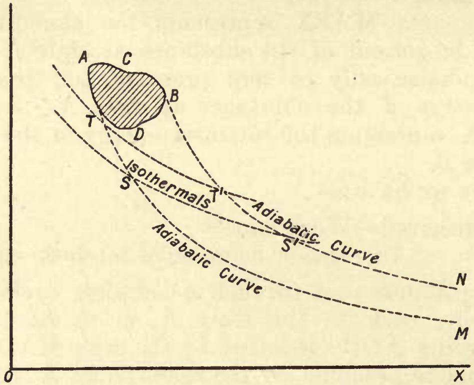

FIG. 1A.

having produced it. Thus in a steam-engine we should not reject heat to the condenser circulating water, but cause it to flow back into the boiler, were it not that more energy would be required to pump the heat back than would be obtained from it when there.

§ 5. **Consideration of Second Law.**—The Second Law of thermo-dynamics was originally propounded by Carnot. He inferred that the efficiency of a perfect engine depended solely on the temperatures between which it worked, and that it was independent of the nature of the substance used as the heat conveyer. He arrived at this conclusion by establishing the important principle that all completely

* An "isothermal curve" of expansion or compression is the curve connecting the pressures and volumes of a gas when a constant weight of it is expanded or compressed at constant temperature.

reversible engines working between the same upper and lower limits of temperature (*i.e.* with the same source of heat and refrigerator) have the same efficiency, no matter what the substance employed. He defined a perfectly reversible engine as being one that, whatever amount of mechanical energy it can derive from a certain thermal agency, if an equal amount be spent in working it backwards an equal reverse thermal effect will be produced. The proof of his principle is, roughly, as follows:—Suppose there are two heat engines, viz.: A of any kind whatever, and B a perfectly reversible engine. Let them be designed to draw the same quantity of heat from the source of heat supply when they are both working forwards. Suppose B be reversed and driven backwards by A, if possible. Then B rejects Q units of heat, say, into the hot body and A receives Q units from the hot body, and there is no transfer of heat from the heat source. If the efficiency of A is greater than that of B working forwards, then U_A is greater than U_B, where U_A and U_B represent the useful work done by A and B respectively. Hence the external work $(U_A - U_B)$ is done by the combination, and this heat transformed into work must be assumed to be derived from the cold body. This process might be continued until all the heat of the cold body or refrigerator is transformed into work. Since this is impossible, it is evident the efficiency of A cannot be greater than that of B. Hence the efficiency of a reversible engine within given limits of temperature is the maximum possible, and therefore all reversible engines working between the same limits of temperature have the same efficiency.

Carnot therefore arrived at the conclusion that the efficiency depended solely on the linear difference of temperatures between which a perfect engine worked—that is

$$\eta = \frac{\text{useful work done}}{\text{heat received}} = \frac{U}{Q} = (T_1 - T_2) \times \text{a constant},$$

where T_1 and T_2 are temperatures measured on some scale.

Hence the η increases as T_2 decreases, and has its maximum value, unity, when $T_2 = 0$.

Then the constant $= \frac{1}{T_1}$, so that

$$\text{Efficiency of a perfectly reversible engine} = \frac{T_1 - T_2}{T_1}.$$

We see, therefore, that the temperature is measured from a point where the substance has no temperature and is absolutely deprived of heat. This point is generally known as the *dynamical zero*, for, could such a low temperature be reached, all molecular motion would cease.

§ 6. **The Dynamical Zero of Temperature.**—Let us consider this dynamical zero of temperature more fully, and see how we can infer its position relative to the thermometric scales in common use. We have seen that at this zero point the substance will have no heat whatever. Let Q_1 and Q_2 be the quantities of heat received by a substance at states 1 and 2. Then if the state 2 represent the dynamical zero, we shall have $Q_2=0$.

Now by the First Law of thermo-dynamics the efficiency of an engine is given by

$$\eta=\frac{Q_1-Q_2}{Q_1} \quad . \quad . \quad . \quad . \quad (1)$$

and by the Second Law it can be stated that

$$\eta=\text{a function of } T_1 \text{ and } T_2 \quad . \quad . \quad . \quad (2)$$

$$\therefore \quad \frac{Q_1}{Q_2}=f(T_1 \cdot T_2) \quad . \quad . \quad . \quad . \quad . \quad (3)$$

Now the heat received by the working substance in an engine depends on four things—its temperature (T), its pressure (p), its volume (v), and the nature of the substance itself (n).

$$\therefore \quad Q_1=F(T_1p_1v_1n_1); \qquad Q_2=F(T_2p_2v_2n_2).$$

$$\therefore \quad \frac{Q_1}{Q_2}=\frac{F(T_1p_1v_1n_1)}{F(T_2p_2v_2n_2)}=\frac{\psi(T_1)}{\psi(T_2)}$$

where $\psi(T_1)$ and $\psi(T_2)$ represent functions of T_1 and T_2. This follows from equation (3), since the ratio of Q_1 to Q_2 depends *only* on temperature.

If $Q_2=0$, then $T_2=0$, and all the heat is converted into work. As one cannot conceive more heat being converted into work than is received, under no conditions can T_2 be negative. We thus get an idea of an absolute zero of temperature, independent of the substance, and corresponding to an absolute deprivation of heat.

To infer the position of this absolute zero relative to a known thermometric scale, say the Fahrenheit scale, con-

sider the expansion of a perfect gas, expanding according to the law

$$pv = t \times \text{constant}$$

between pressures p_1 and p_2 and volumes v_1 and v_2, t being measured on the scale of a perfect gas thermometer.

Then the efficiency of the expansion is given by

$$\eta = \frac{p_1v_1 - p_2v_2}{p_1v_1} = \frac{t_1 - t_2}{t_1}; \quad \text{and } \eta = \frac{Q_1 - Q_2}{Q_1}.$$

$$\therefore \quad \frac{Q_1}{Q_2} = \frac{t_1}{t_2}.$$

Hence

$$\frac{t_1}{t_2} = \frac{T_1}{T_2};$$

that is, our dynamical zero is the same as that of a perfect gas thermometer.

By experiment it is found that if heat be extracted from a gas at constant pressure, its volume decreases proportionally to the decrease of temperature. If we could go on far enough the volume would vanish altogether. For a gas such as air it is found that the volume decreases $\frac{1}{493}$ part for every degree Fahrenheit of decrease of temperature. Hence if the volume of a gas be represented by 493 at 32° F., then

at 31° F. its volume will be 492,
at 0° F. " " 461,
and at -461° F. " " 0.

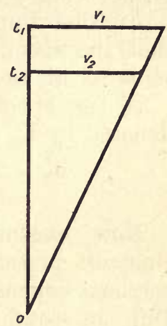

FIG. 2.

In practice, a gas would change its physical state before reaching so low a temperature; but since gases behave nearly as a perfect gas until this change of state is arrived at, we can call this zero (-461° F.) approximately the zero of our perfect gas thermometer, and, therefore, also the absolute dynamical zero.

§ 7. **Laws relating to the Expansion of Gases.**—In the last paragraph it was stated that a perfect gas expands according to the law

$$pv = Ct.$$

This statement is based on two laws, known as *Charles' Law* and *Boyle's Law*.

Charles' Law states that, under constant pressure, equal volumes of different gases increase equally for a similar increment of temperature.

Boyle's Law is the statement that if the temperature of a permanent gas remains constant, and the pressure and volume be varied, the product of the pressure and the volume at any instant is a constant quantity for any given mass of the gas.

Hence $pv = \text{const.}$ when T is constant.
Also by Charles' Law $pa = T$ when v is constant.

Combining these expressions,

$$pv = CT$$

where C is a constant, and T is measured on the scale of the gas thermometer—that is, from the absolute zero of § 6.

C has the value 53·2 for air, and 85·5 for superheated steam, when $p \times v$ is measured in foot-lbs.

Another important law is *Regnault's Law*, which states that the specific heat of a gas at constant pressure is the same for all temperatures.

If the specific heat of a gas at constant pressure be denoted by K_p per lb.—

$$\begin{aligned} K &= 183{\cdot}4 \text{ in foot-lbs. for air;} \\ &= 373 \quad ,, \quad ,, \quad \text{for superheated steam.} \end{aligned}$$

Now, considering the expansion of a perfect gas at constant pressure from a volume v_1 to a volume v_2, evidently the heat expended in external work $= p_1(v_2 - v_1) = C(T_2 - T_1)$. But the total expenditure of heat corresponding to this work is equal to the product of the specific heat into the rise of temperature. Therefore

Total expenditure of heat $= K_p(T_2 - T_1)$;
hence, Heat expended in internal work $= (K - C)(T_2 - T_1)$.

Now, if 1 lb. of gas at constant volume be raised in temperature from T_1 to T_2, the heat absorbed is $K_v(T_2 - T_1)$ where K is the specific heat per lb. at constant volume. Since no external work is done, this also measures the heat required for internal work between the temperatures T_1 and T_2.

$$\therefore \quad (K_p - C)(T_2 - T_1) = K_v(T_2 - T_1);$$
$$\therefore \quad K_v = K - C = 183{\cdot}4 - 53{\cdot}2 = 130{\cdot}2 \text{ foot-lbs. for air.}$$

K_v can be obtained theoretically. It can be shown that

$$\frac{K_p}{K_v}=\gamma$$

where γ is given by

$$v=\sqrt{\frac{\gamma p}{d}},$$

v being the velocity of sound in the gas, d the density, and p the pressure of the gas. K_v has also been found experimentally by Joly, but the experimental determination is difficult. For air, γ has the value 1·408. In general, when a gas expands, its temperature will change; but *Joule* showed that if a perfect gas be allowed to expand without doing any external work and without receiving heat, its temperature will remain constant.

This is not true for air, as its temperature falls $\frac{1}{4}°$ C. for every drop in pressure equivalent to one atmosphere. Advantage is taken of this in some refrigerating machines to obtain a very low temperature by the free expansion of air through a great range of pressure.

§ 8. **Expansion of a Perfect Gas.**—Before proceeding to discuss the actual methods by which work is obtained from gases in heat-engines, it will be advisable to consider how a perfect gas behaves when it expands, and how the heat received and work done during the expansion may be calculated.

The nature of the expansion curve depends on the transfer of heat to other bodies; but in general such expansion curves can be represented by an equation of the form

$$pv^n=\text{constant},$$

when n is given a suitable value.

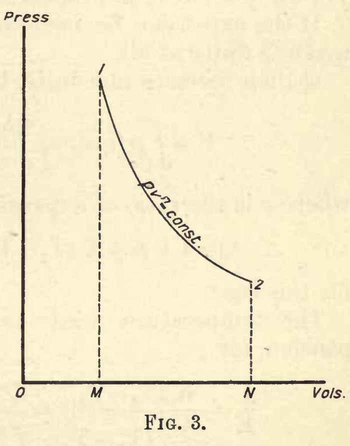

Fig. 3.

Let 1, 2, in fig. 3 be such a curve of expansion. Then the

external work done during expansion from 1 to 2 is given by the area 1, 2, N, M = E.

The internal work done (I) during the same operation is $= K_v(T_2 - T_1)$.

Suppose $T_1 = 0$. Then the quantity of internal work done in raising the gas to a temperature T_2 from absolute zero is equal to $K_v T_2$, and this is defined as the *intrinsic or internal energy* of the gas at the state T_2.

Now, total heat expenditure (Q) from external sources during the expansion 1, 2, is given by

$$Q = I + E.$$

$$I = K_v(T_2 - T_1);\ E = \int_1^2 p\,dv = pv^n\int_1^2 \frac{dv}{v^n} = \frac{p_1v_1 - p_2v_2}{n-1} = \frac{C(T_1 - T_2)}{n-1}.$$

$$\therefore\quad Q = (T_1 - T_2)\left(\frac{C}{n-1} - K_v\right) = K_v(T_1 - T_2)\frac{\gamma - n}{n-1};$$

for $$C = K_p - K_v = K_v(\gamma - 1).$$

If the expansion be *adiabatic*, $Q = 0$;

$$\therefore\quad n = \gamma.$$

For air $\gamma = 1{\cdot}408$, and for steam $\gamma = 1{\cdot}135$.

If the expansion be *isothermal*, $T_1 = T_2$, and hence $n = 1$ to make Q finite at all.

Q then becomes indefinite, but now $pv =$ constant.

$$\therefore\quad E = \int_1^2 p\,dv = pv\int_1^2 \frac{dv}{v} = pv \log_e \frac{v_2}{v_1} = CT_1 \log_e r,$$

where r is the ratio of expansion.

$$\therefore\quad Q = I + E = K_v(T_1 - T_1) + CT_1 \log_e r = CT_1 \log_e r$$

for this case.

The temperature need not necessarily fall during expansion, for

$$\frac{I}{E} = \frac{K_v(T_2 - T_1)}{\frac{C}{n-1}(T_1 - T_2)} = \frac{1-n}{\gamma - 1}, \text{ since } C = K_v(\gamma - 1).$$

During expansion E is necessarily positive.

Hence I is positive so long as n is less than 1, and T_2 is greater than T_1; that is, the temperature of a gas increases during expansion if $n<1$. Correspondingly, if n be greater than 1, then also T_2 is less than T_1; *i.e.* the temperature falls during expansion.

Evidently when $n=1$, then $I=0$, and $T_2=T_1$; that is, the expansion is isothermal.

CHAPTER II.

HOT-AIR ENGINES.

§ 9. **General Considerations.**—In the last paragraph of the preceding chapter the method was discussed by which the work done by a perfect gas for a given heat supply could be calculated. Practically all heat-engines use gases or vapours, whose properties approach those of a *perfect gas*, as the means of converting heat into useful work. The working substance receives heat from the heat source, expands in a cylinder doing work, and finally rejects any heat unused into a cold body or condenser. The working fluid is then ready for a fresh supply of heat from the heat source, so it finally returns to the same state in which it started—that is to say it passes through a *closed cycle* of operations. There are thus three essentials for a heat-engine: (1) a hot body; (2) a cold body; (3) the working fluid.

One of the most convenient working fluids to use is *air*, and the present chapter will be devoted to describing some types of heat-engines that employ this gas for their working substance. In different engines the expansive fluid passes through quite different cycles of operations. It is intended here to consider only four types of cycles that may be adopted, as, if the methods of calculating the heat accounts of these cycles be fully appreciated, the reader will have no difficulty in dealing with other types that may be met with.

§ 10. **Carnot's Cycle for a Heat-Engine.**—Carnot was the first to suggest a cycle for the working fluid of a heat-engine, though, owing to mechanical difficulties, no actual engine was constructed using his cycle of operations. His proposed cycle was as follows:—Suppose a gas is in the

condition p_1v_1 in a cylinder, and occupying the space 01′ in the figure. Apply a hot body A to it, and let the piston move from 1′ to 2′. The curve of expansion of the gas will then be 12, and evidently this will be an isothermal curve of expansion at the temperature of the hot body, as A is supposed in contact with the gas while the piston is travelling from 1′ to 2′. When the piston has reached the point 2′ suppose A removed, and then the gas will continue to expand adiabatically, since no heat is being received, until 3′ is reached. Now apply the cold body B and let the piston

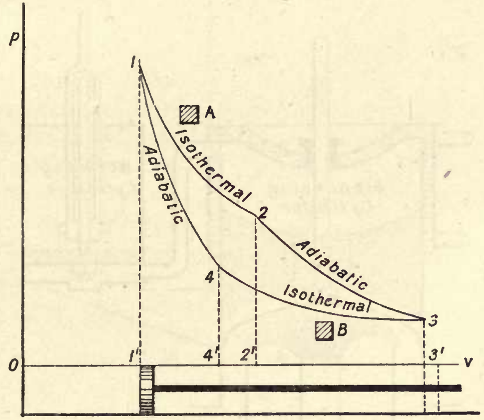

FIG. 4.—Carnot's Cycle.

return, compressing the gas in the cylinder to 4′. Evidently the compression will again be isothermal, but at the temperature of B. Finally at 4′ remove B, and the compression will be completed adiabatically until the piston is at the end of its stroke. Consider now the distribution of heat during the cycle, and let suffices refer to the figures in the diagram. Then

Heat received from 1 to $2 = Q_{12} = CT_1 \log_e r$ per lb. of air, where r is the ratio of isothermal expansion.
Heat received from 2 to $3 = Q_{23} = 0$.
Heat rejected from 3 to $4 = R_{34} = CT_3 \log_e r'$.
Heat rejected from 4 to $1 = R_{41} = 0$.

It can readily be shown $r = r'$ in order that the final adiabatic curve must pass through the starting-point, 1.

$\therefore$ Heat converted into useful work

$$=\mathrm{U}=\Sigma\mathrm{Q}-\Sigma\mathrm{R}=\mathrm{C}(\mathrm{T}_1-\mathrm{T}_3)\log_e r.$$

Hence, efficiency of a Carnot's cycle

$$=\eta=\frac{\mathrm{U}}{\Sigma\mathrm{Q}}=\frac{\mathrm{C}(\mathrm{T}_1-\mathrm{T}_3)\log_e r}{\mathrm{CT}_1\log_e r}=\frac{\mathrm{T}_1-\mathrm{T}_3}{\mathrm{T}_1};$$

as before, in § 5.

§ 11. **Stirling's Engine.**—The first practical hot-air engine was invented by the Rev. Robert Stirling in 1816.

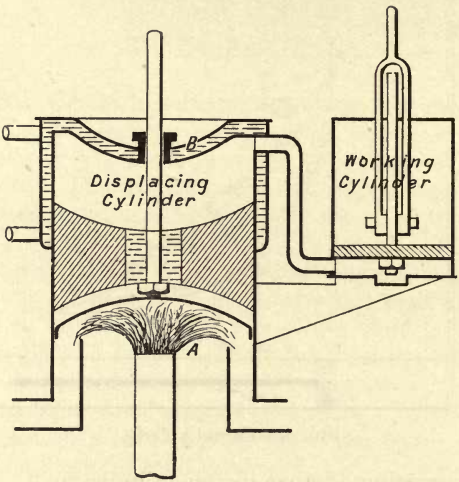

FIG. 5.—Section of Stirling's Engine.

He employed isothermal expansion during the whole forward stroke of the working piston; then, by means of a large separate cylinder containing a non-conducting displacing piston worked off the engine shaft, he was able to suddenly cool the working fluid from the temperature of the heat source to that of the cold body. The latter consisted of a water jacket enclosing the top of the displacing cylinder. A sudden change of temperature and pressure at constant volume was thus obtained. The *working piston* on its return stroke compressed the air in the cylinder isothermally, at the temperature of the water jacket; and at the end of its stroke the displacing piston again moved suddenly up, forcing the

air from the water-jacket side of it to the hot side, thus causing a sudden rise of pressure at constant volume. His cycle was, therefore, as shown in fig. 6.

12 represents isothermal expansion of the air in contact with the hot body A.

$$\therefore \quad Q_{12} = CT_1 \log_e r \text{ per lb. air.}$$

23 represents fall of temperature and pressure at constant volume, heat being rejected to B.

$$\therefore \quad R_{23} = K_v(T_1 - T_3);$$

K_v being the specific heat of air at constant volume.

34 represents isothermal compression by the working piston, the air being in contact with the cold body B, *i.e.* the cooling water.

$$\therefore \quad R_{34} = CT_3 \log_e r.$$

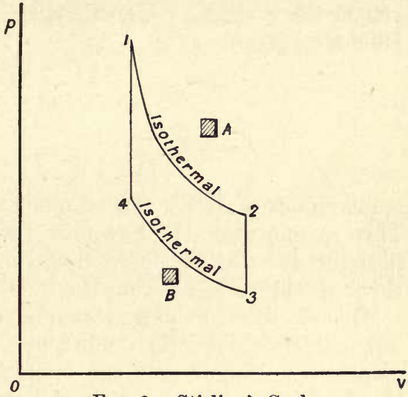

FIG. 6.—Stirling's Cycle.

41 represents rise of temperature at constant volume, heat being received from the hot body A.

$$\therefore \quad Q_{41} = K(T_1 - T_3).$$

$\therefore$ Total expenditure of heat $= \Sigma Q = CT_1 \log_e r + K_v(T_1 - T_3)$,

and Total heat rejected $= \Sigma R = CT_3 \log_e r + K_v(T_1 - T_3)$.

$$\therefore \quad U = \Sigma Q - \Sigma R = C(T_1 - T_3) \log_e r;$$

$$\eta = \frac{U}{\Sigma Q} = \frac{C(T_1 - T_3)\log_e r}{CT_1 \log_e r + K(T_1 - T_3)}.$$

Evidently the efficiency of this engine is less than that of a Carnot's cycle, since the term $K_v(T_1 - T_3)$ occurs in the denominator.

This term may be reduced by fitting what is known as a *regenerator* to the displacing piston. This consists of a number of perforated metal sheets through which the air has to pass on its way from one side of the displacing piston to the other. As the hot air passes through the regenerator it gives up its heat, and so less heat is rejected to the cooling

2

water; and when the displacing piston forces the cold air back to the hot surface, it receives back the heat previously given to the regenerator, and so less heat is taken from the hot body to bring the air back to the higher temperature. If the efficiency of the regenerator be e, the heat now taken from the hot body during rise of temperature will be only $(1-e)K_v(T_1-T_3)$, as the remainder will be received from the regenerator itself. Hence, with a regenerator fitted, the efficiency is

$$\eta = \frac{C(T_1-T_3)\log_e r}{CT_1 \log_e r + (1-e)K_v(T_1-T_3)}$$

$$= \frac{T_1-T_3}{T_1}$$

approximately; for e can be made as much as ·9 in practice. The regenerator is shown in the centre of the displacing plunger in the sectional diagram, the air actually passing through the plunger from the hot body to the cold body.

This engine is evidently nearly perfect thermo-dynamically, but mechanically it is inefficient. The motion of the large displacing plunger, and the energy to effect compression by the working piston, is derived from a heavy fly-wheel fitted to the engine shaft; also for high efficiency, since the lower temperature T_3 cannot be much below that of the atmosphere, the upper temperature must be considerable. The bottom of the displacing cylinder, therefore, must be kept very hot, and consequently the cylinder bottom becomes rapidly oxidised and requires frequent repair. For these reasons Stirling's engine is only suitable for purposes requiring little power.

§ 12. **Ericcson's Engine.**—This engine is represented diagramatically in fig. 7. Its principal feature consisted in the fact that the working cylinder had the heat directly applied to one of its ends, whilst the compression of the air did not occur in the working cylinder but was performed by a separate pump worked off the engine shaft. A regenerator was fitted as in the Stirling engine, and the admission, etc., of air to the cylinders was regulated by valves actuated at correct times by cams on the engine shaft.

Consider the cycle of operations (as represented by fig. 8) through which the air passed during one revolution of the engine.

During A3 air is drawn in from the atmosphere by the pump, and during the next stroke of the pump it is compressed isothermally by the aid of a water jacket round the pump barrel, as represented by 34, and then expelled into a receiver as indicated by 4B. The air then passes through the regenerator, as shown in fig. 7, and into the working cylinder at the pressure of the receiver. This is represented by the line B1. In the cylinder the air expands isothermally from 1 to 2, the air being in contact with the heat source.

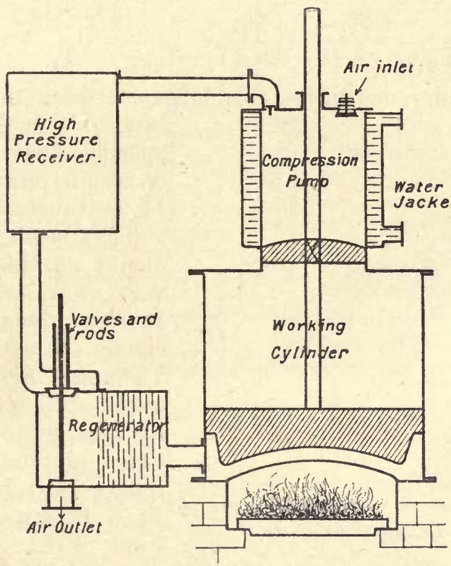

FIG. 7.—Section of Ericcson's Engine.

Finally, the air is exhausted through the regenerator to the atmosphere, represented by 2A. Hence

B12A is the complete diagram for the working cylinder;
A34B is the complete diagram for the pump;
∴ 4123 is the nett diagram of work available.

In this cycle it should be noticed that change of temperature takes place at constant pressure, not at constant volume as in Stirling's cycle.

Consider now the transfer of heat.

$$Q_{41} = K_p(T_1 - T_3) \text{ per lb. of air;}$$
$$Q_{12} = CT_1 \log_e r;$$
$$R_{23} = K_p(T_1 - T_3);$$
$$R_{34} = CT_3 \log_e r.$$

$$\therefore \eta = \frac{U}{\Sigma Q} = \frac{\Sigma Q - \Sigma R}{\Sigma Q} = \frac{C(T_1 - T_3) \log_e r}{CT_1 \log_e r + K_p(T_1 - T_3)}.$$

If a regenerator be fitted, we can write as before—

$$\eta = \frac{C(T_1 - T_3) \log_e r}{CT_1 \log_e r + (1 - e)K_p(T_1 - T_3)} = \frac{T_1 - T_3}{T_1} \text{ nearly.}$$

The only difference in the calculation between this and the last cycle is that K_p, the specific heat of air at constant pressure, must be used instead of K_v.

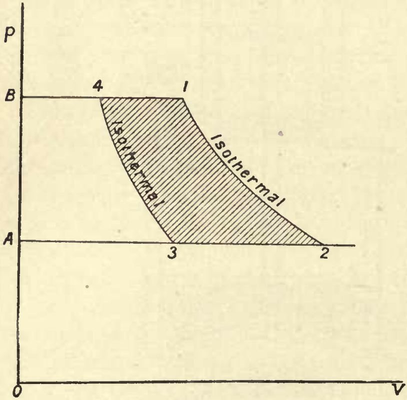

Fig. 8.—Ericcson's Cycle.

The mechanical efficiency of this engine is very low, though its thermo-dynamic efficiency is considerable. The vessel *Ericcson*, fitted out in 1853, was propelled by this form of engine. She had four giant cylinders 14 ft. in diameter by 6 ft. stroke, but the total I.H.P. was only about 300. The fuel was only 1·87 lbs. per I.H.P. per hour, but the mechanical efficiency of the engines was very small.

§ 13. **Buckett's Engine.**—The practical objections to all air-engines are that—

(1) They are very bulky for their power.
(2) The mechanical efficiency is small.
(3) Lubrication of the cylinders is difficult owing to the high temperature used.

In addition, the engines that have been just described have the disadvantages of—

(4) The difficulty of supplying an efficient heating surface.
(5) The rapid burning away of this heating surface.

These last two objections can be overcome by employing the products of combustion of the heating fuel as the

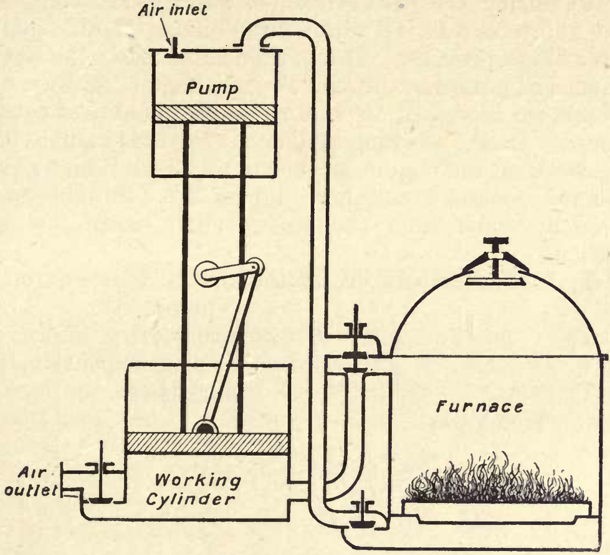

FIG. 9.—Section of Buckett's Engine.

working fluid, in the cylinder itself. Such engines are called *internal combustion engines*, in contradistinction to those that receive their heat through a surface.

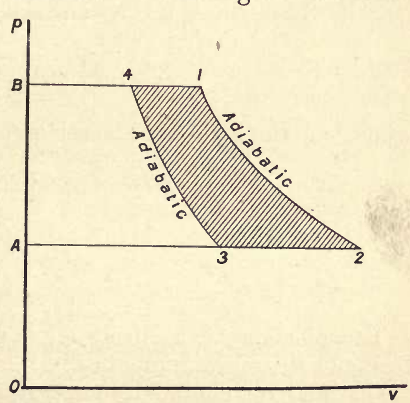

FIG. 10.—Buckett's or Joule's Cycle.

Buckett's engine is an engine of this type, employing the furnace gases as the working fluid. The principle of this engine will be understood from the annexed diagrammatic section, and the cycle of operations shown in fig. 10. Buckett adopted what is known as *Joule's cycle* of operations. A separate pump worked

off the engine shaft drew in air from the atmosphere during A3, and compressed it adiabatically, as shown by 34 (fig. 10), during the next stroke, *no* water jacket being fitted to the pump-barrel. 4B represents delivery of air from the pump at high pressure. It was then admitted to the working cylinder (B1), passing through the furnace on its way. The air therefore increased in volume to the point 1 at constant pressure. In the working cylinder it expanded adiabatically doing work on the engine, as shown by 12, and finally it was exhausted to the atmosphere during 2A. Suitable valves, worked by cams from the engine shaft, controlled these operations.

If T_4 be the temperature of the air on delivery from the pump,

T_1 „ „ „ furnace,

T_2 „ „ „ air after expansion,

and T_3 „ „ „ atmosphere,

evidently (see § 8)

$$Q_{41}=K_p(T_1-T_4)\,;\ Q_{13}=0$$
$$R_{23}=K_p(T_2-T_3)\,;\ R_{34}=0.$$

$$\therefore\ \eta=\frac{U}{\Sigma Q}=\frac{\Sigma Q-\Sigma R}{\Sigma Q}=\frac{K_p(T_1-T_4-T_2+T_3)}{K_p\,(T_1-T_4)}=1-\frac{T_2-T_3}{T_1-T_4}.$$

To find T_2, we have

$$\frac{T_1}{T_2}=\frac{p_1v_1}{p_2v_2}\ \text{and}\ p_1v_1{}^{\gamma}=p_2v_2{}^{\gamma}.$$

$$\therefore\ \frac{T_1}{T_2}=r^{\gamma-1}$$

where r is the ratio of adiabatic expansion.

So also $\dfrac{T_4}{T_3}=r^{\gamma-1}$. $\therefore\ \dfrac{T_1}{T_2}=\dfrac{T_4}{T_3}$.

$$\therefore\ \eta=1-\frac{T_2-T_3}{T_1-T_4}=1-\frac{T_2\left(1-\frac{T_4}{T_1}\right)}{T_1-T_4}=1-\frac{T_2}{T_1}=\frac{T_1-T_2}{T_1}.$$

This efficiency is again less than that of a Carnot cycle, for T_2 is not the *lowest* temperature.

Gas and oil engines are also internal combustion engines, but their consideration will be deferred to the next chapter.

CHAPTER III.

GAS AND OIL ENGINES.

§ 14. **Gas-Engines. General Considerations.**—As stated in the last chapter, gas and oil engines are internal combustion engines. They possess the advantage of combining considerable mechanical efficiency with a high thermo-dynamic efficiency.

In gas-engines the system now usually adopted is to draw a mixture of gas and air into a cylinder, then to make the working piston compress this mixture, and, when the piston is just at the end of its stroke, to ignite the mixture at constant volume by means of an electric spark, a jet of flame, or a red-hot ignition tube. This causes an explosion of the gas, and the rise of pressure does work on the piston during the next stroke.

The proportions of gas to air forming the explosive mixture can vary from 1 to 4, up to 1 to 14 by volume, and even more if the mixture be previously heated. At atmospheric temperature and pressure ordinary coal-gas will not form an explosive mixture with air, outside these limits.

The pressure produced on explosion will depend on this richness of mixture, on the amount of compression to which the mixture is subjected before ignition, and on the initial chemical composition of the gases employed.

Diagram 11 shows some results of the effect of richness of charge obtained by Mr. Clerk when exploding different mixtures of Glasgow coal-gas and air, at atmospheric pressure, in a vessel of constant volume.

The highest temperature obtained was with the richest mixture, and was about 1920° C. The curves clearly show

that combustion is gradual, and, under these conditions, may be controlled by dilution of the mixture. By experimenting with mixtures of constant composition at different original pressures he also showed that the explosion pressures for any mixture are proportional to the pressures before ignition.

These experiments show approximately what occurs in the cylinder of a gas-engine on ignition, when the piston is at the compression end of its stroke. Actually the gas does not all combine at once, but combustion continues after the explosion, causing the expansion curve to rise above the adiabatic curve during the working stroke. This effect is known as *after burning*.

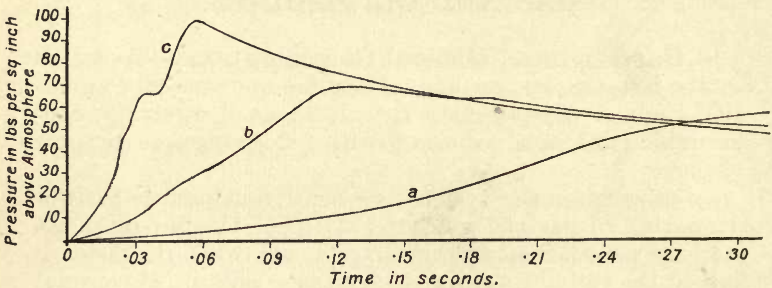

FIG. 11.—Explosion Pressures at Constant Volume.

a—Mixture of	1 volume	of gas	to	13	vols.	of air.
b	,,	1	,,	,,	9	,,
c	,,	1	,,	,,	5	,,

For practical reasons a water jacket has to be fitted to the cylinders of gas-engines to prevent the temperature of the cylinder walls becoming too high for proper lubrication. Hence, though theoretically it is better to have complete immediate combustion to obtain the maximum range of temperature, yet in practice it is found advantageous to have delayed combustion to reduce the initial temperature and consequent loss of heat to the water jacket, and also because it maintains a uniformly higher pressure throughout the working stroke.

Even with "after burning" it is found that as much as one-third to one-half of the total heat of combustion is lost to the water jacket. In practice, about half the calorific value of the gas is given out at ignition, and the remainder during the working stroke.

The effect of the water jacket is to counteract "after burning" as far as the expansion curve is concerned, for it absorbs the additional heat generated. Hence the actual curve of expansion of the burnt gas approaches the adiabatic curve.

§ 15. **Thermo-dynamic Cycles used in Gas-Engines.** —Gas-engines may be divided into two classes, according as the gaseous mixture employed in the cylinders is compressed or not before ignition.

The class of engines employing *no compression* may be further subdivided into two types: the first directly using the force of the explosion to drive the piston; while the second type uses the atmospheric pressure and weight of piston, etc., as the driving force, the explosion merely lifting the piston with great velocity while it is disconnected from the driving mechanism.

The first type is represented by Lenoir's engine * of 1860. This was the first gas-engine to be brought into practical use, and employed the following cycle of operations: air and gas in suitable proportions were drawn into the engine cylinder, at atmospheric pressure, during a portion of the forward stroke; cut-off then occurred, and the mixture was ignited by an electric spark. The pressure rose quickly, followed by expansion of the burning gases, work being done on the piston for the remainder of the stroke. On the return stroke the piston drove out the products of combustion at atmospheric pressure.

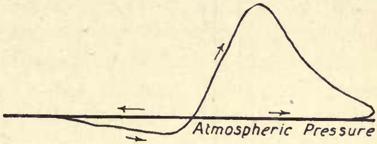

FIG. 12.—Diagram from Lenoir Engine.

The actual indicator diagram of the Lenoir engine was as shown in fig. 12.

The engine was double acting, two explosions occurring per revolution. A water jacket was fitted to prevent excessive heating of the cylinder. Principally for the reason that the explosive mixture was not compressed before ignition, the heat removed by the water jacket bore a large proportion to the total heat generated; on this account, and also because of the limited final range of expansion of the burnt gases, the efficiency of this engine was low.

* For details of this and other gas and oil engines, see W. Robinson's book on *Gas and Petroleum Engines*; also Bryan Donkin's *Gas Engines*.

The Lenoir engine used about 95 cubic feet of gas per I.H.P. per hour, or about four to five times as much as a modern gas-engine.

The second type of non-compression engine was constructed in a practical form by Otto and Langen. Fig. 13 shows the indicator diagram obtained with this engine. The explosive mixture is admitted at atmospheric pressure for a portion of the stroke; it is then exploded, and the piston, which is quite free, is driven up like a projectile. As the piston recoils it is geared up to the driving shaft, and does useful

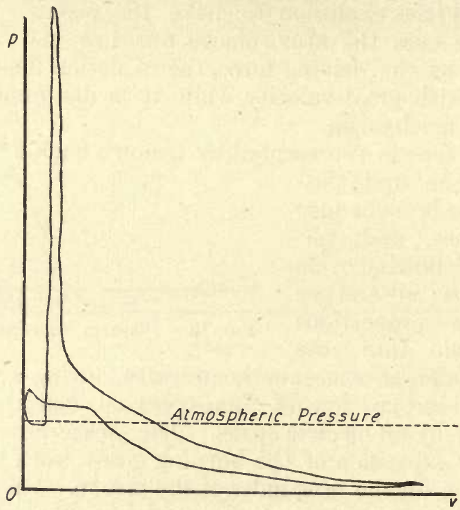

FIG. 13.—Diagram from Otto and Langen Engine.

work as it slowly falls owing to its own weight, the gases in the cylinder meanwhile expanding to a pressure below that of the atmosphere.

Other atmospheric engines of this type have been made by Halliwell, Robson, and others.

Gas-engines in which the explosive mixture is *compressed* before ignition may also be divided into two types—namely, those in which the ignition is followed by a sudden rise of pressure; and secondly, those in which the compressed gases, after having been ignited, burn slowly at constant pressure.

The advantage of compressing the explosive mixture before ignition in order to make the subsequent expansion larger,

appears to have been first clearly recognised by Beau de Rochas in a French patent in the year 1862.

He defined the desirable conditions for a gas-engine to be the greatest cylinder volume combined with the least cooling surface; the greatest rapidity of action of the piston; the greatest possible expansion of the gaseous mixture; and the greatest possible pressure at the commencement of expansion.

He proposed the cycle of operations shown in fig. 14, the complete cycle to take place on one side of the working piston during two revolutions of the crank-shaft—that is, during four strokes of the piston.

In this cycle the gaseous mixture is drawn in during the whole of one stroke of the piston, as represented by A3 in the diagram, and then compressed adiabatically during the return stroke (as shown at 34) into the clearance volume. It is then ignited at the constant volume of the clearance, which causes a sudden increase of pressure during 41, with subsequent adiabatic expansion during the third stroke, 12. Finally, the exhaust valve is opened by mechanical means causing a drop in pressure during 23, and the burnt gases are expelled by the piston at atmospheric pressure during the fourth stroke, 3A.

The first successful gas-engine using this cycle was patented in 1876 by Dr. Otto, and since then this cycle has been used by the majority of modern gas-engine patentees. It is commonly known, however, as the *Otto cycle.* The theoretical efficiency of the cycle can be readily calculated on the assumption that the curves of compression and expansion are truly adiabatic.

For if Q be the heat due to explosion,

$$Q = K_v(T_1 - T_4)$$

where K_v is the specific heat of the mixture, and T_1 and T_4 are the temperatures at the points 1 and 4 of the diagram.

Also, the work done *by* the gas in expanding = E = area

$$123A = K_v(T_1 - T_2);$$

and the work done *on* the gas in compressing it = R = area

$$43A = K_v(T_4 - T_3).$$

$\therefore$ Useful work done by the engine = U = E − R

$$= K_v(T_1 - T_2 - T_4 + T_3).$$

$$\therefore \text{ The efficiency} = \frac{U}{Q} = \frac{T_1 - T_2 - T_4 + T_3}{T_1 - T_4}$$

$$= 1 - \frac{T_2}{T_1} \text{ since } \frac{T_1}{T_2} = \frac{T_4}{T_3};$$

$$= 1 - \frac{1}{r^{\gamma - 1}}$$

where r is the ratio of adiabatic expansion (see § 13).

Hence the efficiency of such a cycle depends only on the ratio of expansion or compression, and is greater the larger this ratio.

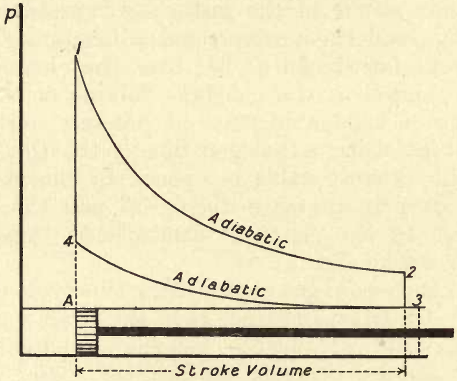

FIG. 14.—Otto Cycle.

As stated previously, the amount Q in practice is only about half the total heat of combustion owing to loss of heat to the water jacket.

The number of modern gas-engines using approximately the Otto cycle is considerable, and no attempt will be made to enumerate them; but in order to show the type of indicator diagram obtained from such engines in practice, fig. 15 has been drawn. This shows an indicator card taken from the well-known Otto-Crossley type of engine, and for purposes of comparison it has been drawn to the same scale of pressures and stroke volumes as the theoretical diagram of fig. 14.

That the efficiency of such engines is actually increased by increasing the amount of compression, in agreement with

theoretical considerations, is strikingly shown by the following table, published by the Society of Arts.

Date.	Type of Engine.	Diameter of Cylinder in inches.	Stroke in inches.	Compression Pressure in lbs. per square inch above Atmosphere.	Indicated Efficiency.	Actual Efficiency.	Gas per I.H.P. in cubic feet.
1882–8	Otto-Crossley	9	18	38	·33	·17	24
1888–94	,,	9·5	18	61·6	·40	·21	20·5
1895	,,	7	15	87·5	·428	·25	14·8

The type of engine in which the compressed gases burn slowly at constant pressure without explosion, is seldom met with at the present time. The Brayton engine of 1872 is representative of the principle involved; and the Diesel oil-engine uses a similar cycle of operations, but a description of its method of working will be given later.

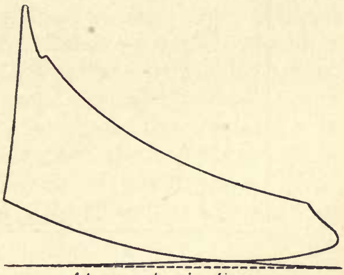

FIG 15.—Indicator Diagram from Otto-Crossley Engine.

§ 16. **Causes of Loss of Thermal Efficiency of Gas-Engines.**—The principal causes of loss of efficiency of gas-engines are :—

(1) All the heat is not supplied at the maximum temperature.

(2) The water jacket conveys away a considerable portion of the heat of combustion of the gases.

(3) The exhaust gases are discharged at a high temperature and pressure.

(4) Owing to clearance spaces, burnt gases are left in the cylinder after exhaust and dilute the fresh charge.

(5) Frictional losses occur owing to wire drawing during admission and expulsion of the gases.

The reasons for these losses and the means of reducing them are as follows :—The first loss appears inevitable. As

regards the second, a certain loss is a practical necessity, but the amount will depend on the excess of the temperature of the working fluid over the temperature of the cylinder walls; the extent of the surface of the cylinder walls exposed per lb. of working fluid; and the periodic time of the engine cycle. Now, the surface per lb. of fluid varies inversely as the cylinder diameter; hence for the same power, gas-engines should have cylinders with large diameter and short stroke. Also, the periodic time of the engine should be small, and hence the revolutions per minute should be large, for the heat transference to the water jacket per cycle will vary inversely as the revolutions for a given temperature difference.

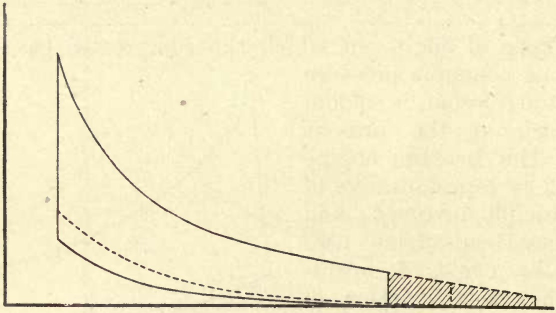

FIG. 16.—Effects of Increasing Expansion.

The third source of loss is due to sudden release, and can be reduced by increased expansion; but this necessitates increased compression in the Otto cycle, and the latter is limited by practical considerations. Increased expansion can be obtained without a corresponding increase of compression by adding on a toe to the Otto cycle diagram, as shown in fig. 16, but the increased frictional losses due to the greater stroke of engine then necessary would probably more than balance any gain of work thus obtained.

It must be borne in mind that increasing expansion and compression in engines using the Otto cycle increase the maximum pressure and working stresses, and require a heavier engine for a given power, and also increase the mean temperature of the cylinder walls; so that the extra loss to the water jacket may more than balance the increase of efficiency due to the increase of expansion.

The fourth cause of loss is due to the unrejected products

of combustion mixing with the incoming fresh charge of gas and air, thus reducing the efficiency of combustion not only by dilution but also by preventing the full quantity of pure air being present during the succeeding stroke. To increase the efficiency of gas-engines it is desirable, therefore, to sweep out the burnt gas from the clearance space at the termination of the exhaust stroke. This operation is known as *scavenging*, and can be effected in several different ways. The burnt clearance gas can be withdrawn by a separate pump or other suitable means; but Messrs. Crossley have adopted an ingenious method by which the same result is effected without mechanical means. This is the invention of Mr. Atkinson, and simply consists in fitting the engines with a long exhaust pipe (about 65 feet long), whilst the air admission valve is opened just before the gas-valve and a little before the end of the piston stroke. The exhaust gases are discharged through the exhaust pipe with a rather violent puff, and the energy of their motion in the long pipe causes the puff to be followed by a partial vacuum in the cylinder clearance space. Hence when the air-valve is opened there is a rush of pure air into the clearance space, which sweeps out before it the residue of burnt gases. This invention has resulted in a marked gain in the efficiency of such engines.

The fifth source of loss is a necessity, and cannot be entirely prevented.

§ 17. **Gaseous Fuel.**—The gas used for gas-engines may be ordinary coal-gas obtained from the destructive distillation of coal, and, for small powers, its convenience often compensates for its comparatively great cost. The composition of coal-gas will vary considerably with the kind of coal, and with the conditions under which the gas is made. Mr. L. T. Wright gives the following percentage composition by volume as a sample of gas distilled from Newcastle coal:—

Hydrogen	67·1	per cent.
Marsh gas	22·6	,,
Carbon monoxide	06·1	,,
Carbon dioxide	01·5	,,
Heavy hydrocarbons	01·8	,,
Nitrogen	00·8	,,
Sulphuretted hydrogen	00·1	,,

1 lb. of coal may be taken as yielding about 4·5 cub. ft. of coal-gas. The total heat evolved in the complete combustion of 1 cub. ft. of such gas will obviously depend on its chemical constituents; but for ordinary coal-gas the heat of combustion is not far short of 600 to 650 British thermal units per cubic foot, when the products of combustion pass off at about 60° F.

In the calculation of the heat of combustion of a given gas in an engine cylinder, or the heat rejected during exhaust, it is necessary to know the specific heats of the mixture of gas and air both before and after explosion. These specific heats will depend on the composition of the gas and the specific heats of its constituents, on the quantity of air present during combustion, and on the temperature. However, for approximate calculations the following values may be used:—

Substance.	Specific Heat at Constant Pressure in B.T.U.	Specific Heat at Constant Volume in B.T.U.
Ordinary coal-gas . . .	·25	·18
Air	·237	·168
Burnt gases . . . . . (Air burnt with gas in the proportion of 12 : 1 by weight.)	·27	·198

For gas-engines of large power a cheap form of gas is necessary for economical working. One typical method of producing such gas is that introduced by Mr. Dowson, though many other methods are now in vogue. Dowson gas is made by passing a mixture of steam and air through a red-hot mass of coke or anthracite. By this means the steam is decomposed into its constituent gases, and the resulting gas has an approximate composition given by:— Hydrogen 19 per cent., Carbon monoxide 25 per cent., Carbon dioxide 6 per cent., and Nitrogen 50 per cent. by volume.

The gas thus obtained has only one quarter the calorific value of ordinary coal-gas, but by restricting the proportion of air admitted to the engine cylinder, and compressing highly, it is found to ignite well. Such gas plants are very compact, require little attention, and have many advantages.

§ 18. **Oil Fuel.**—Oil can be used as the motive fluid for engines in a similar manner to gas. Some oils are more volatile than others, and the methods of obtaining mechanical energy from them differ according to their properties.

The methods that can be employed are:—

(*a*) Fuel such as coal or coke can be used to evaporate naphtha or benzine, which, from their low latent heat and temperature of evaporation, require a small boiler only, and the vapour can then be used similarly to steam in a steam-engine cylinder.

(*b*) The oil can be converted into a true gas at a high temperature, and the gas used as in an ordinary gas-engine.

(*c*) The oil can be evaporated at a low temperature, and the vapour then mixed with air and exploded in an engine cylinder; or it may even be pumped direct into the engine cylinder and exploded with air without previous evaporation.

(*d*) Liquid oil can be sprayed into the furnace of an ordinary boiler through nozzles, by the aid of jets of steam or air, or simply by a force-pump.

The so-called "spirit oils" are naphtha, benzine, and petroleum, having a flash-point lower than 73° F. and a density of ·7 to ·73. The oils used for petrol engines have flash-points from 73° F. to 120° F., with densities from ·73 to ·82.

The method (*a*) is not now used in practice; but Messrs. Yarrow have made engines for river launches on this principle, with economical results.

The methods (*b*) and (*c*) are the ones adopted by practically all the makers of modern oil-engines. No detailed description of the various types of oil-engines can be given here; but it may be noticed that the Hornsby-Ackroyd engine is typical of method (*b*), while the Priestman and Diesel engines illustrate method (*c*). Practically all oil-engines employ the *Otto cycle* for their thermo-dynamic cycle of operations, and so in no way differ thermo-dynamically in their working from the gas-engines already described.

The Diesel engine may, however, be noticed briefly, as its cycle differs from the Otto, and the engine in its modern form has attained a very high efficiency. The Diesel engine employs the cycle of operations shown by the indicator diagram of fig. 17. In this engine, air is highly compressed more or less

adiabatically (about 35 atmospheres per sq. in.) in the engine cylinder before any oil is admitted at all. At the end of the piston stroke the oil is injected into the cylinder by means of a separate pump and air reservoir, and immediately burns owing to the high temperature of the compressed air in the cylinder. *No explosion* occurs, but the oil *burns* at constant pressure as it is pumped in during a portion of the next forward stroke of the piston. The supply of fuel is then cut off and a prolonged expansion (more or less adiabatic) ensues, until release occurs at the end of the stroke.

In a thermo-dynamic sense, it is a defect of gas and oil engines that combustion usually begins while the working substance is comparatively cold. As just described, the Diesel engine overcomes this defect by endeavouring to

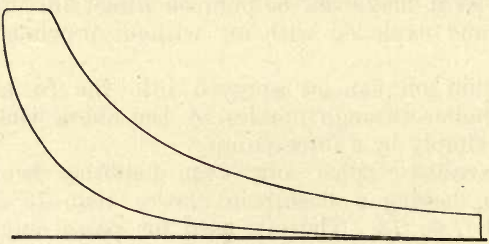

FIG. 17.—Diesel Engine Cycle.

reach the temperature of combustion before combustion commences, by compressing the air in the cylinder to a very high degree before the fuel is admitted.

Professor Meyer in 1900, testing a 30-H.P. Diesel engine, obtained the following results at normal load using American petroleum oil:—

I.H.P. 39·52; Oil per I.H.P. per hour, ·343 lbs.
B.H.P. 30·17; Oil per B.H.P. per hour, ·449 lbs.

Allowing for the work done by the air-pump in compressing the air of injection, he found that 38 per cent. of the heat contained in the oil was converted into indicated work—a very high efficiency.

The method (*d*) of burning oil fuel is now being largely introduced, especially for marine boilers. By this means mechanical energy can readily be obtained from those crude, heavy oils, which cannot be used in the cylinders of ordinary

oil-engines. The method adopted is to spray the oil through suitable burners on to the top of a bed of live coal in a boiler furnace, or else simply to spray and burn the oil by itself. Various forms of sprayers are in use, but the principle consists in forcing air or steam at high pressure through a form of ejector which draws in the oil and discharges it into the boiler furnace. The oil must be heated up to its flash-point by passing it through a heater containing steam coils to render it readily combustible. The flash-points of the oils used vary from 180° F. to 270° F., and it is advisable to heat the oils above these points to get smokeless combustion. It is very important, however, not to heat the oil too highly, as then the lighter constituents in the oil (which can never be entirely eliminated even from the heaviest oils) split up into their elements and deposit pure carbon in the form of flakes and particles, and these rapidly choke any valves or pipes through which the oil may pass. For this reason it is inadvisable to heat any oil to be used for burning over 100° above its flash-point, and even then suitable oil-strainers should be fitted to prevent the small nozzles of the burners becoming choked with sooty particles. Air should be drawn in uniformly all round the oil-jets, and to get smokeless results it is very important to secure a good and gradual admixture of the oil and air at a high temperature, and also that the jet of oil should not impinge upon any brickwork or cold substance in the furnace. By forcing much air into the furnace, the combustion can be made smokeless in any case; but this is dilution only, and will not be attended by good evaporative results. From the results of a number of trials it appears that about 13·5 lbs. of water evaporated per lb. of oil from and at 212° F. is about the result that may be expected from oil burnt in this way, as compared with about 10·5 lbs. when coal only is used. If burning well, no residue is left by the oil either in the furnaces or on the boiler tubes. When steam is used as the spraying medium, the burners appear to use about 4 per cent. of the steam generated by the boiler in which they are being used.

For marine work, the advantages over coal of this type of fuel are:—

(1) Great convenience and reduction of labour owing to the cleanliness of burning.

(2) Great convenience in stowage and manipulation and in the power of any boiler being readily adjustable; and

(3) The larger calorific value of oil than coal for a given weight.

Its disadvantages are:—

(1) The large quantities of black smoke formed unless considerable care is taken.

(2) The loss of fresh water when steam injection is used.

(3) The deafening noise produced when steam burners are used.

(4) The relatively greater cost of oil than coal owing to its somewhat limited supply.

The second and third objections have been overcome by using what are known as *pressure burners.* In these no steam or air jet is used, but the oil is forced into the furnaces through nozzles by an oil pressure pump at a pressure which varies from 300 to 100 lbs. per square inch, according to the design. As in the case of the steam burners the oil must be previously heated to from 0° F. to 50° F. above its flash-point, and the air supply for properly burning the oil must be gradually supplied and thoroughly mixed with the oil to cause complete combustion.

CHAPTER IV.

REFRIGERATING MACHINES.

§ 19. **Conditions for Refrigeration.**—Refrigerators are the direct opposite of heat-engines, and may, in fact, be considered as reversed heat-engines. Their object is to cool matter to a lower temperature than that of the surrounding medium. Heat must therefore be withdrawn from a cold body and rejected into a hot body, by the expenditure of mechanical work. In § 5 we have seen that in a heat-engine, in order to get the maximum amount of work out of the heat received, under given conditions, it is necessary that:—

(*a*) The working substance must at every point be at the same temperature as the source of heat when receiving heat, and at that of the cold body when rejecting it.

(*b*) The working pressure must always be balanced by the resistance.

If these conditions are complied with, the cycle of operations is said to be *reversible.* It is unnecessary for the working substance to receive all its heat at the highest available temperature and reject it at the lowest, in order that the cycle may be reversible. For a particular reversible cycle, however, the *maximum efficiency* is obtained when this is the case, as is obvious from fig. 18, where the area between the isothermals TT′ and SS′ is evidently greater than the area ABCD, which will represent the work done by any less efficient engine working between the same limits of temperature.

We thus see thermo-dynamic reversibility is the criterion of perfection for a heat-engine, and the same holds for a refrigerator, the refrigerating effect being greater for a given

expenditure of work the more nearly the refrigerating machine approaches perfect reversibility.

In a heat-engine we have seen

$$Q - R = U$$

where Q = heat supplied from the hot body;
R = heat rejected to a cold body;
U = useful work done by the engine.

In a refrigerating engine we have,

$$R' = Q' - U'$$

where R' = heat supplied by the cooling body;
Q' = heat rejected into the hot body, generally condensing water;
U' = work done on the engine by external sources.

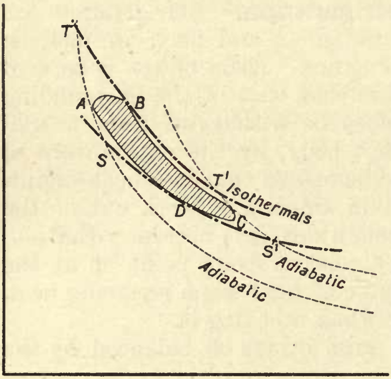

Fig. 18.

In the case of heat-engines we are concerned with the ratio $\frac{U}{Q}$, that is, the amount of work that can be obtained from a given supply of heat.

In refrigerators the cooling effect will depend on the ratio of R' to U', that is, the amount of heat that can be withdrawn from the cold body for a given expenditure of work. The ratio $\frac{R'}{U'}$ is therefore very important, and is generally called the *coefficient of performance* of the machine.

Apply these considerations to the Carnot cycle shown in § 10. Suppose, instead of receiving heat from A, we go round the cycle in the opposite direction and compress adiabatically during 32 and isothermally during 21. Heat will then be rejected to A. Now let the working substance expand, first adiabatically during 14, and finally isothermally at the temperature of B during 43. To keep this expansion isothermal, heat must be received from B. Hence by the expenditure of an amount of mechanical work equivalent to the area 1234, heat can be removed from B and transferred

into A. B will thus get colder and colder, so that by reversing the cycle of operations required for a perfect heat-engine, we see how a refrigerating effect can be obtained. The resulting effects of working forwards or backwards round the cycle can be tabulated thus:—

Transfer of Heat.	Operation.	Working forwards as for Heat-Engine.	Operation.	Working backwards as for Refrigerator.
$CT_1 \log_e r$ . .	12	From A.	21	Into A.
0 . .	23 ↓	From A.	32 ↑	Into A.
$CT_3 \log_e r$ . .	34	Into B.	43	From B.
0 . .	41	Into B.	14	From B.

It is evident from the above table that Carnot's cycle is exactly reversible, and, therefore, as suitable for a refrigerator as a heat-engine.

The "coefficient of performance" is given by

$$\frac{R'}{U'} = \frac{CT_3 \log_e r}{CT_1 \log_e r - CT_3 \log_e r} = \frac{T_3}{T_1 - T_3}.$$

§ 20. **Refrigerators using Non-reversible Cycles.**— Many refrigerating machines use cycles in which only certain operations are entirely reversible. In such cases we must find the quantities R′, Q′, and U′ all separately, and cannot get such a simple relation between R′ and U′ as for the Carnot cycle in the last paragraph.

In 1862 Dr. Kirk invented a refrigerator, using air as the working fluid, and employing a reversed Stirling's cycle of operations with a regenerator fitted (see § 11).

Comparing the transfer of heat between the hot and cold bodies in the two cases, a table can be written down as before:—

Transfer of Heat.	Operation.	Working forwards as a Heat-Engine.	Operation.	Working backwards as a Refrigerator.
$CT_1 \log_e r$ . .	12	From A.	21	Into A.
$K_v(T_1 - T_3)$.	23 ↓	Into B.	32 ↑	From A.
$CT_3 \log_e r$ . .	34	Into B.	43	From B.
$K_v(T_1 - T_3)$.	41	From A.	14	Into B.

Here, evidently, operations 23 and 41 are not exactly reversible; for, as in a heat-engine, heat is rejected to B during 23; when the cycle is reversed, heat should be taken *from* B during 32 and not from A, and similarly for the operation 41.

The heat taken from B during each cycle in the Kirk's refrigerator is evidently given by

$$R' = CT_3 \log_e r - K_v(T_1 - T_3),$$

which is less than that for a Carnot cycle working between the same limits of temperature. With a regenerator of efficiency e fitted, however, we can write

$$R' = CT_3 \log_e r - (1-e)K_v(T_1 - T_3);$$

so that, with a very efficient regenerator, Kirk's machine approaches very closely the thermo-dynamic efficiency of the Carnot cycle.

The coefficient of performance is then given by

$$\frac{R'}{U'} = \frac{CT_3 \log_e r - (1-e)K_v(T_1 - T_3)}{C(T_1 - T_3)\log_e r}.$$

The reason for the loss of efficiency in the Stirling cycle is evident; for the heat rejected to produce change of temperature at the end of the piston stroke is lost, while in the Carnot cycle all the heat is received at the higher and rejected at the lower temperature.

This example shows once again that non-reversible engines are not so efficient as those that are completely reversible, and also that for maximum efficiency there must be no transfer of heat during change of temperature.

Rankine and Lord Kelvin, about the year 1852, proposed the following cycle for refrigerators:—Compress air adiabatically, cool it, then expand it again adiabatically to a low temperature, and then bring it into contact with the cold body. As will be seen from fig. 19, the cycle is simply a reversed Joule's cycle.

The Bell-Coleman refrigerating machine, invented about 1877, worked on this principle, and in more recent years the well-known Haslam and Hall air-machines have been designed employing the same cycle of operations.

The accompanying figure represents diagrammatically the various operations performed :—

A1 represents the suction stroke of the compression pump from the atmosphere.

12 is approximately adiabatic compression; the air then passing into the cooler or condenser.

23 represents the reduction in volume of the air at constant pressure due to the cooling effect of the condenser.

30 shows the adiabatic expansion of the cooled air in the expansion cylinder, the temperature here falling to a very low degree, say to T_0.

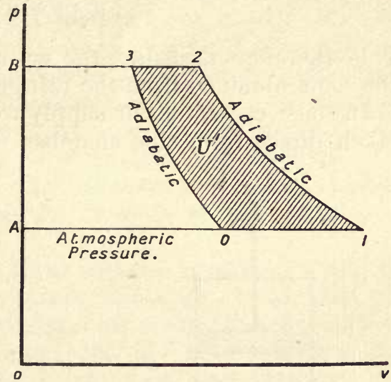

FIG. 19.

Let T_1 be the temperature of the atmosphere, T_2 the temperature at the end of compression, and T_3 the temperature of the condensing water.

$$\therefore \quad Q'_{23} = K_p(T_2 - T_3)\,; \quad R'_{01} = K_p(T_1 - T_0)\,;$$
$$Q'_{30} = 0\,; \qquad R'_{12} = 0.$$

$$\therefore \quad U' = Q' - R' = K_p(T_2 - T_3 + T_0 - T_1).$$

$\therefore$ Coefficient of performance =

$$\frac{R'}{U'} = \frac{T_1 - T_0}{T_2 - T_3 + T_0 - T_1} = \frac{1}{\dfrac{T_2 - T_3}{T_1 - T_0} - 1} = \frac{T_0}{T_3 - T_0}\,;$$

for

$$\frac{T_2}{T_1} = \frac{T_3}{T_0} = r^{\gamma - 1}$$

where r is the ratio of adiabatic expansion (see § 13). Approximately also $T_3 = T_1$, the atmospheric temperature.

Suppose the cold air at temperature T_0 be required to

cool a room—for meat storage, for instance. Let T_4 be the average temperature of such a room. Then, evidently,

Heat abstracted from cold chamber per lb. of air is

$$=K_p(T_4-T_0).$$

This therefore measures the actual nett cooling effect on the room obtained from the refrigerator.

In some cases, the air supply to the compression pump is taken direct from this chamber instead of from the outside

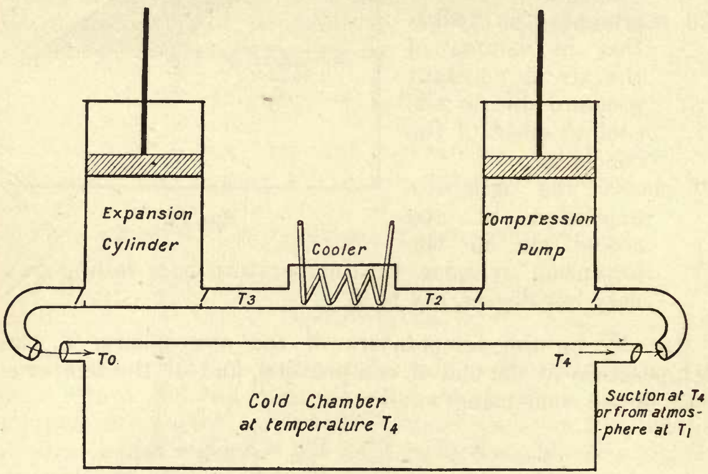

FIG. 20.—Diagram of Refrigerator using Cycle of Fig. 19.

atmosphere. T_1 then becomes equal to T_4, and the *same* air continually circulates from the machine to the cold room, and back to the machine again.

The amount of heat extracted per stroke will be the same as before; but, owing to the weight of air dealt with per stroke being greater at the lower temperature, the performance of the machine will be increased.

Another practical advantage of this method is due to the fact that a certain quantity of moisture is always present in the atmosphere. Hence snow and ice slowly form in the cylinders and on the valves of refrigerators; and thus when only a given quantity of air is introduced into the various

chambers, a much smaller quantity of snow and ice is formed in a given time.

§ 21. **Vapour Refrigerators.**—Any vapour can be used instead of air as the working fluid for refrigerators, provided the pressures it gives at the temperatures of the hot and cold bodies are suitable.

Water vapour is quite unsuitable for refrigerating purposes. Its vapour pressure is considerably less than 1 lb absolute per sq. in. even at the highest temperature likely to be reached on the warm side of a refrigerating machine.

Sulphuric ether has been tried as a working substance, but its vapour pressure is below the pressure of the atmosphere for temperatures under 95° F.

Hence to use either of these vapours great care would be necessary in the fitting of glands, joints, etc., to prevent air leaking into the refrigerator and destroying the efficiency of the process. Moreover, owing to the low pressures the volume per lb. of vapour is very great, and therefore the machines would necessarily be very bulky, which renders them unsuitable for modern requirements.

Sulphurous acid has been employed by M. Pictet as a refrigerating vapour. It has a vapour pressure of about $4\frac{1}{2}$ atmospheres per sq. in. at about 70° F., and so is suitable for such machines. Its thermal performance is greater than for air; but the liability of this vapour to form sulphuric acid in the presence of air, with consequent corrosive effects on metal, renders the use of this vapour somewhat limited.

Anhydrous ammonia is a vapour in considerable use for refrigeration. At normal temperatures the range of pressure obtained in ammonia machines is from about 20 to 170 lbs. per sq. in. Its thermo-dynamic performance compared to air is large, and the size of a machine for a given refrigerating effect is well within practical limits, so that ammonia forms a very suitable vapour in every way.

The disadvantages to its use are—(*a*) that it actively dissolves brass and copper, so that these materials cannot be used in the construction of these machines; (*b*) leakage allows very noxious fumes to escape; (*c*) special stores are required to be carried on board ship to replenish losses, which is not necessary with an air plant.

Carbon dioxide is another vapour largely used for refrigerators. It is least bulky of any of the vapours, but its

pressures are correspondingly higher, reaching as much as 1040 lbs. per sq. in. at about 68° F. Hence carbonic acid refrigerators, though very compact, must be strongly made, and special precautions must be taken to prevent leakage.

§ 22. **Thermo-dynamic Cycles of Vapour Refrigerators.**—The most perfect cycle an ideal vapour refrigerator could use would be a reversed Carnot cycle. Under these conditions, the cycle of operations would be as shown in fig. 21. For vapours, isothermal lines are lines of constant pressure at the temperatures corresponding to those pressures, so that, following round the cycle, 21 will represent adiabatic compression; 14, rejection of heat at constant pressure at the upper temperature, *i.e.* the temperature of the cooling water; 43, adiabatic expansion; and 32, the absorption of heat from the cold body, again at constant pressure and temperature. At the point 2, after contact with the cold body, the vapour as a rule is not dry but contains a certain proportion of moisture, say q_2 of its mass. During the adiabatic compression 21 this moisture evaporates, and if the proportion of moisture q_2 is suitably arranged, the vapour will be just dry at the end of compression. This need not necessarily be the case, as it is quite possible to arrange that the vapour shall have a dryness fraction q_1, say, at the point 1, or even be superheated. Assuming it has a dryness given by q_1 at the point 1 of the cycle, the amount of heat rejected to the cooling water during 14 will be q_1L_1 where L_1 is the latent heat of the vapour at this temperature, for the vapour condenses to a liquid during the operation of cooling at the higher pressure. As before, we can tabulate the transfer of heat for a *perfect cycle* for a vapour and compare it with that of a vapour heat-engine:—

Transfer of Heat.	Operation.	As a Vapour Heat-Engine.	Operation.	As a Vapour Refrigerator.
0 . .	12	...	21	...
$(q_2-q_3)L_2$.	23 ↓	Into B.	32 ↑	From B.
0 . .	34 ↓	...	43 ↑	...
q_1L_1 . .	41	From A.	14	Into A.

The above cycle for a vapour is evidently completely reversible.

In practice, the adiabatic expansion 43 cannot be obtained by making the condensed vapour do work in a cylinder. Partial re-evaporation can be actually carried out by passing the liquid through a restricted orifice on its way to the cold body; but no work is obtainable by this process, so that in actual vapour refrigerators the work up to the adiabatic expansion curve on the perfect cycle diagram must be supplied from external sources.

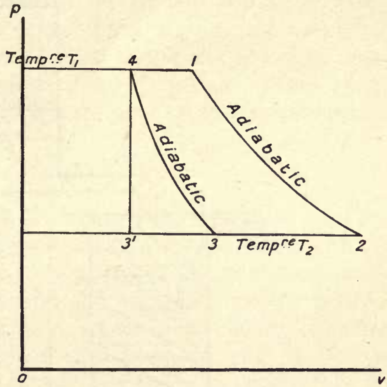

FIG. 21.—Cycles of Vapour Refrigerators.

The most perfect cycle for a vapour obtainable in actual practice is *Rankine's cycle.* This cycle is reversible except for the part 43′ (fig. 21), which represents the fall of temperature of the condensed vapour with consequent rejection of heat to the *cold* body.

Rankine's cycle may be therefore tabulated thus:—

Transfer of Heat.	Operation.	As a Vapour Heat-Engine.	Operation.	As a Vapour Refrigerator.
0	12	...	21	...
q_2L_2	23′ ↓	Into B.	3′2 ↑	From B.
$h_1 - h_2$	3′4	From A.	43′	Into B.
q_1L_1	41	From A.	14	Into A.

where h_1 is the sensible heat at the temperature t_1, etc. Now let us apply these principles to examine the working of the ordinary ammonia or carbonic acid refrigerating machines.

The type of plant employed generally consists of a compression pump, which condenses the vapour; a condenser, which causes the vapour to become liquid at the temperature of the condensing water; and an evaporator—as shown diagrammatically in fig. 22. This evaporator is connected to the condenser, and a regulating valve is fitted to adjust the

drop of pressure between them. Partial evaporation takes

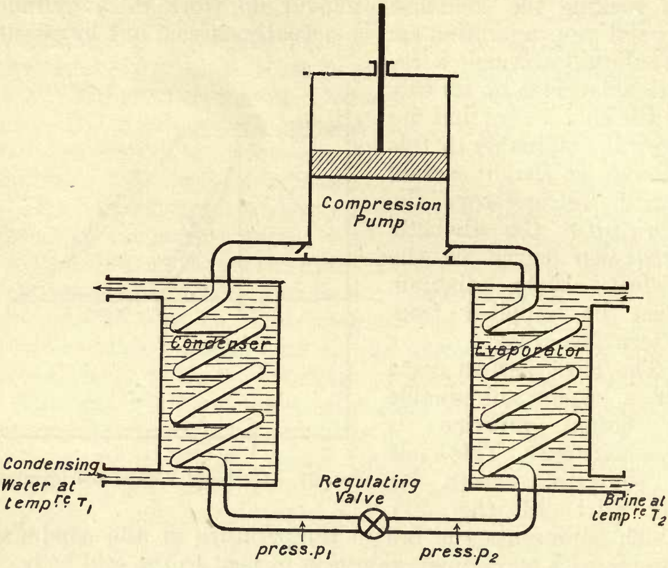

FIG. 22.—Section of Vapour Refrigerator.

place at this valve and the remainder in the evaporator coils. The evaporator coils are surrounded by brine, which is thus reduced to a very low temperature and can be carried off to cool chambers, or to freeze water, etc.

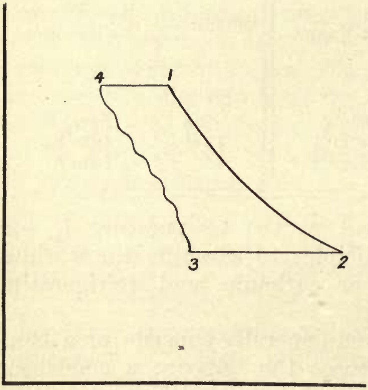

FIG. 22A.

The expansion at the regulating valve is found to be irregular, and is represented in fig. 22A by the wavy line 43. Evidently the dryness fraction of the vapour at the point 3 will be given by

$$q_3 L_2 = h_1 - h_2,$$

where h_1 and h_2 are the sensible heats per lb. at the upper and lower temperatures.

The heat taken out of the cold brine per cycle is

$$R' = (q_2 - q_3)L_2$$

per lb. of vapour, while the work done by the pump, neglecting the small volume of the condensed liquid itself, is the area 123′4—*not* the area 1234, since the work 433′ does not reappear as work on the compression pump. The heat rejected to the condensing water per cycle is evidently

$$Q' = q_1 L_1$$

per lb. of vapour.

The dryness fraction q_1 will depend on the state of the vapour at the point 2 before compression commences. If q_2 be considerable, the vapour at the upper temperature will not be dry after compression, and the process of refrigeration is then said to be *wet*. For a suitable q_2, however, the vapour will be just dry after compression, and if q_2 be greater than this the vapour may be even superheated at the upper temperature. The refrigeration is then said to be *dry*. The value of q_2 can be altered in practice by adjusting the regulating valve, and on the proper adjustment of this valve will depend the refrigerating effect for any given limits of temperature.

§ 23. **Refrigerating Effect dependent on Heat Properties of Vapours.**—Since $R' = (q_2 - q_3)L_2 = q_2L_2 - (h_1 - h_2)$, it is evident the refrigerating effect produced depends on the ratio of q_2L_2 to $(h_1 - h_2)$ for any particular vapour.

Hence it is important to choose a vapour in which L_2, that is its latent heat, is large compared to its specific heat.

The numerical values of the latent heats and specific heats of various substances, in thermal units, is given in the following table:—

	Latent Heat at 32° F. in B. T. Units.	Liquid Specific Heat in B. T. Units.
Water	1092	1
Sulphurous acid	164	·33
Ammonia	568	·9
Carbonic acid	100	·53

Obviously the ratio of L_2 to $(h_1 - h_2)$ is greatest for water;

but for reasons already discussed, water vapour cannot be used for refrigerators.

Ammonia is the next best, and this forms another argument in favour of ammonia machines.

If temperatures be plotted on a heat base, the curves of specific and latent heats for ammonia and carbonic acid will be as shown in fig. 23.

We observe that at 88° F. the latent heat of carbonic acid is zero. This is called its *critical temperature*, for above this

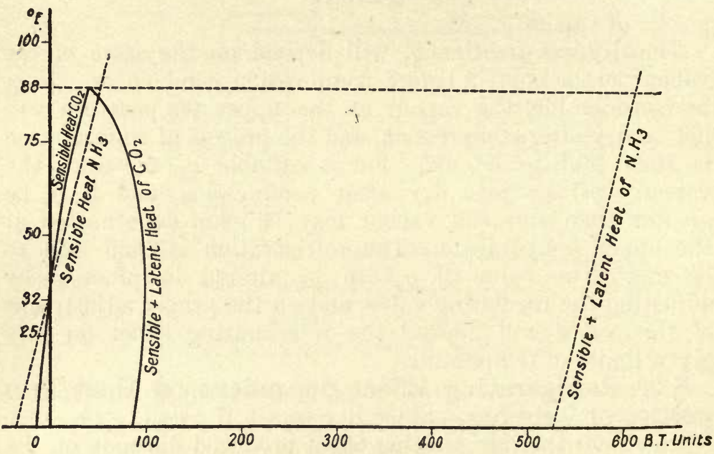

FIG. 23.—Curves showing Relative Values of Sensible and Latent Heats from 32° F. of Ammonia and Carbonic Acid.

temperature no amount of compression and condensation will condense the gaseous CO_2 into a liquid form. Hence a carbonic acid refrigerator working between this upper limit of temperature and any lower limit of temperature will no longer refrigerate unless the action of the machine radically alters, for no evaporation can take place to produce a refrigerating effect. These CO_2 refrigerators do work, however, under these conditions, the explanation being somewhat as follows:—If air expands from 101 to 1 atmosphere doing no work on external bodies, its temperature will fall 25° C., and this same result obtains when the vapour pressure of carbonic acid is allowed to fall under similar conditions. At the regulating valve in the carbonic

acid machines the pressure drop can be regulated, and the temperature on the low-pressure side made to fall to a point much below that of the brine in the evaporator. The vapour thus entering the evaporator coils absorbs heat from the brine by virtue of its specific heat, and so refrigerates.

This principle is made use of in Hampson and Linde's machine for producing liquid air. Air is compressed to 200 atmospheres per sq. in. and then allowed to fall to atmospheric pressure by passing through a valve. It therefore drops in temperature some 50° C., and is then led to an air-jacket surrounding the air-pipe leading to the expansion valve. The next supply of air at 200 atmospheres pressure thus reaches the expansion valve some 50° C. below atmospheric temperature, and when this second quantity is expanded it falls a further 50° C. It is then led to the jacket again, and by this compound principle a very low temperature can be reached—sufficiently low to liquefy air.

CHAPTER V.

THE TRANSMISSION OF POWER BY COMPRESSED AIR.

§ 24. **Plant required for Air-Power Transmission.**—Compressed air forms a convenient medium by which mechanical energy can be transmitted through considerable distances without excessive losses.

The machinery necessary consists of—a prime mover or engine to drive the compressing mechanism; an air-pump in which the air is compressed; a reservoir in which to store the high-pressure air after compression; piping; and a motor to convert the potential energy of the air back into mechanical work.

§ 25. **Losses of Work in Air Compressor and Motor.**—In an ideal air-compressing plant the work necessary to be done by the prime mover in compressing air between given limits of pressure p_2 and p_3 and discharging it at the higher pressure would be that represented by the area 1234 in fig. 24, where 23 is an isothermal curve. If there were no losses in the pipes or air-motor, this area would represent the potential energy of the compressed air which could be obtained from the air-motor as mechanical energy by making it pass through the operations 4321 in the reverse direction.

However, for the curve 23 to be isothermal it would be necessary for the compression in the pump, and the expansion in the motor, to be carried out very slowly, in order that the necessary amount of heat to keep the curve isothermal might be taken from and given to the cylinder walls of the pump barrel and motor cylinder respectively.

Practically compression and expansion are carried out too quickly to be isothermal, and so the cycle of operations of the air-compressor is not exactly repeated in the air-motor.

If compression and expansion be carried out very quickly the time is too short for any sensible interchange of heat with the cylinder walls, and hence the curves of compression and expansion will approximate to adiabatic curves, as shown at 2A and 3B in fig. 24.

The work necessary to be done in compressing the air is thus given by the area 12A4 in practice, while the energy obtainable from the air-motor is represented by the area 43B1; for the air in transmission will cool down till its temperature at 3 is the same as that of the surrounding atmosphere, which is at the temperature T_2. There is thus a loss of work in the transaction represented by the area 2A3B. We observe this area is similar to the Joule's cycle of fig. 10, and so we can readily calculate the extent of this loss. Evidently

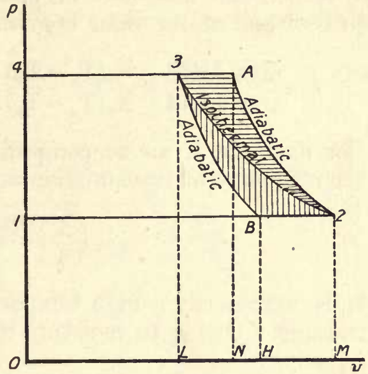

FIG. 24.—Showing Losses in Air Compression and Expansion.

Loss of compression $= L_C =$ area 2A3;

$= $ area M2AN $+$ area NA3L $-$ area M23L;

$= K_v(T_A - T_2) + p_3(V_A - V_3) - CT_2 \log_e r;$

$= K_p(T_A - T_2) - CT_2 \log_e r$

where r is the ratio of isothermal compression, *i.e.* $\frac{V_2}{V_3}$;

for $T_3 = T_2$; $p_3 V_A = CT_A$ and $p_3 V_3 = CT_3$; and $K_p = K_v + C$.

So also in the motor—

Loss by expansion $= L_E =$ area B32;

$=$ area M23L $-$ area B3LH $-$ area M2BH;

$= CT_2 \log_e r - K_v(T_3 - T_B) - p_2(V_2 - V_B);$

$= CT_2 \log_e r - K_p(T_2 - T_B);$

$\therefore$ Total loss $= L_C + L_E = K_p(T_A + T_B - 2T_2)$.

Now since 2A and 3B are adiabatic curves between the

same limits of pressure, it follows, as has already been shown (§ 13), that

$$\frac{T_A}{T_2} = \frac{T_3}{T_B} = R^{\gamma-1}$$

where R is the ratio of adiabatic compression and $\gamma = 1{\cdot}408$ for air.

Hence the efficiency of power transmission due to compressor and motor alone is given by

$$\eta = \frac{\text{area } 1B34}{\text{area } 12A4} = \frac{K_p(T_3 - T_B)}{K_p(T_A - T_2)} = \frac{T_B}{T_2} = \frac{T_B}{T_3} = \frac{1}{R^{\gamma-1}} = \left(\frac{p_2}{p_A}\right)^{\frac{\gamma-1}{\gamma}}.$$

For example, if air be compressed from 1 to 4 atmospheres, the efficiency of transmission is given by

$$\eta = \left(\frac{1}{4}\right)^{\frac{1{\cdot}408-1}{1{\cdot}408}} = {\cdot}67.$$

T_A is necessarily a high temperature, but T_B is often below freezing. Owing to moisture in the air passing through the plant, this low temperature would in time prevent the air-motor working due to the formation of ice in the expansion cylinder. To prevent this the air is sometimes heated before entering the expansion cylinder by means of steam jets, or some form of "preheater."

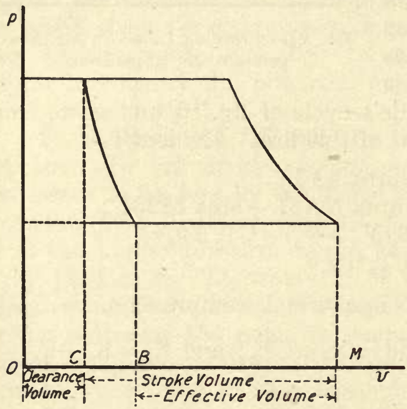

FIG. 25.—Effect of Clearance.

Clearance in the cylinders is also prejudicial to the efficiency of compressing plant. This is seen in fig. 25 by the air in the clearance space expanding from OC to OB, due to its own elasticity on the suction stroke. Hence a smaller quantity of atmospheric air is drawn in, and the pump must take more strokes to deal with the same quantity of air, with a resulting greater loss by friction.

§ 26. **Method of Reducing Losses during Compression and Expansion.**—As has been pointed out, if the curves of compression and expansion in compressor and motor can be made isothermal, the energy of compression can be made to reappear entirely during expansion in the air-motor, and then the efficiency of the plant will be unity. In practice the compression cannot be strictly isothermal, owing to the speed of the operation; and, in fact, if no precautions are taken, it will approximate more nearly to adiabatic.

To prevent as far as possible the temperature of the air rising during compression, water may be injected into the compressing cylinder with the air, and in this way the curve of compression which would be represented by

$$pv = \text{constant}$$

if *isothermal*, and by

$$pv^{1 \cdot 408} = \text{constant}$$

if *adiabatic*, may be made to lie between the two. The curve will then be roughly expressed by

$$pv^{1 \cdot 2} = \text{constant}.$$

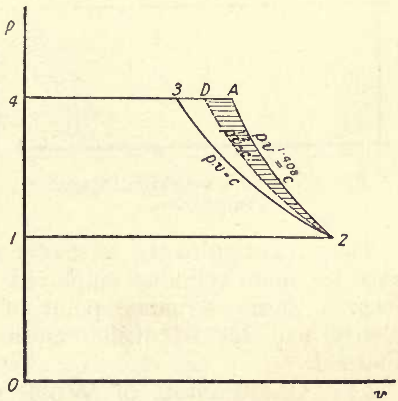

Fig. 26.—Work saved in Compression with Water Injection.

The saving in work thus obtained will be represented by the shaded area 2AD of fig. 26.

A considerable saving in the waste of power may also be obtained by *staging* the compression and expansion that is carrying out the total range of compression and expansion by increments in several successive cylinders, the air being cooled between each stage.

The effect of doing this in three stages is shown by fig. 27, where 23 represents the approximately adiabatic compression in the first stage, 34 represents the cooling effect before the air is compressed again adiabatically in the second cylinder during 45, and 67 represents the compression in the final cylinder to the required final pressure. The air

being cooled down to atmospheric temperature between each stage, the points 2461 will all lie on one isothermal; and the saving of work by staging the adiabatic compression instead of doing all the work in *one* cylinder will be represented by the shaded area A34567.

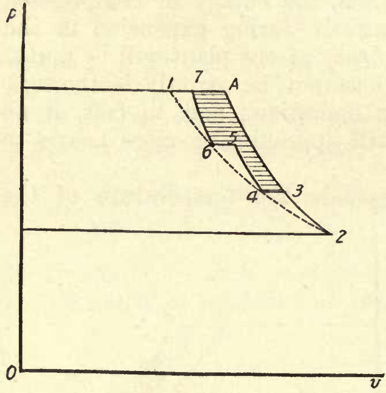

FIG. 27.—Work saved by "Stage" Compression.

Water may also be injected into each cylinder during compression, and thus the curves 23, 45, 67, etc., may each be made to approach the isothermal more nearly than the adiabatic, with a resulting gain of work. These principles are actually carried out in such air-compressors as Belliss', etc.

The expansion in the air-motor may be staged in the same way, the more cylinders employed the greater being the gain from a thermo-dynamic point of view, as then the more nearly will the expansion curve approximate to the isothermal.

§ 27. **Calculation of Work done during Compression and Expansion.** 1. *Single-stage Compression and Expansion.*—Suppose air be drawn in at a pressure p_1 and absolute temperature T_1, and compressed to a final pressure p_2 according to the law

$$pV^n = \text{constant},$$

and then discharged to a reservoir at the higher pressure.

The work to be done will be represented by the area 12A4 of fig. 24.

Now $$p_1 V_1^n = p_2 V_2^n;$$

$$\therefore \quad \frac{V_2}{V_1} = \left(\frac{p_1}{p_2}\right)^{\frac{1}{n}}.$$

∴ Work of compression and discharge per lb. of air

$$=\mathrm{U}=\frac{n}{n-1}(p_2\mathrm{V}_2-p_1\mathrm{V}_1);$$

$$=\mathrm{CT}_1\cdot\frac{n}{n-1}\left(\frac{p_2\mathrm{V}_2}{p_1\mathrm{V}_1}-1\right) \text{ for } p_1\mathrm{V}_1=\mathrm{CT}_1;$$

$$=\mathrm{CT}_1\cdot\frac{n}{n-1}\left\{\left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}}-1\right\}.$$

After leaving the compressor the air cools down in the reservoir to the atmospheric temperature T_1, and the volume

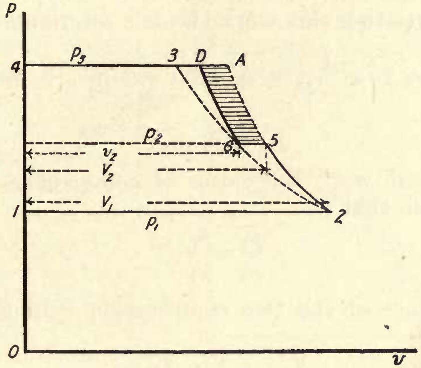

FIG. 28.—Two-stage Compression.

V_2 per lb. becomes, say, only v_2 at the same pressure p_2. Hence when this air is made to do work in a motor, we get—

Work done by motor per lb. of air

$$=\mathrm{U}'=\frac{n}{n-1}(p_2v_2-p_1v_1);$$

$$=\mathrm{CT}_1\cdot\frac{n}{n-1}\left(1-\frac{p_1v_1}{p_2v_2}\right) \text{ for } p_2v_2=\mathrm{CT}_1;$$

$$=\frac{\mathrm{CT}_1.n}{n-1}\left\{1-\left(\frac{p_1}{p_2}\right)^{\frac{n-1}{n}}\right\}.$$

And the efficiency of the plant $=\dfrac{\mathrm{U}'}{\mathrm{U}}$.

2. *Two-stage Compression and Expansion.*—In this case the work of compression and discharge will be represented by the area 1256D4 in fig. 28, where p_3 is the final pressure

of air required, and p_2 is the pressure at the end of the first stage of compression. The work area can evidently be represented by the sum of two terms similar to that found for a single-stage compressor. Thus—

Work of compression and discharge per lb. of air

$$=\mathrm{U}=\frac{\mathrm{CT}_1.n}{n-1}\left\{\left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}}-1\right\}+\frac{\mathrm{CT}_1 n}{n-1}\left\{\left(\frac{p_3}{p_2}\right)^{\frac{n-1}{n}}-1\right\}$$

$$=\frac{\mathrm{CT}_1.n}{n-1}\left\{\left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}}+\left(\frac{p_3}{p_2}\right)^{\frac{n-1}{n}}-2\right\}.$$

By differentiation this work (U) is a minimum when

$$p_1^{\frac{1-n}{n}}\cdot\frac{n-1}{n}\cdot p_2^{-\frac{1}{n}}+p_3^{\frac{n-1}{n}}\cdot\frac{1-n}{n}\cdot p_2^{\frac{1-2n}{n}}=0,$$

or

$$p_2=\sqrt{p_1 p_3}.$$

Hence for least work the ratios of compression in the two stages are such that

$$\frac{p_3}{p_2}=\frac{p_2}{p_1};$$

and the volumes of the two compression cylinders are then given by

$$\frac{\mathrm{V}_1}{v_2}=\frac{p_2}{p_1}=\sqrt{\frac{p_3}{p_1}}.$$

Also, since

$$\frac{\mathrm{T}_2}{\mathrm{T}_1}=\frac{p_2\mathrm{V}_2}{p_1\mathrm{V}_1}=\left(\frac{p_2}{p_1}\right)^{\frac{n-1}{n}}\text{ and }\frac{\mathrm{T}_3}{\mathrm{T}_1}=\left(\frac{p_3}{p_2}\right)^{\frac{n-1}{n}},$$

$$\therefore\quad\frac{\mathrm{T}_2}{\mathrm{T}_1}=\frac{\mathrm{T}_3}{\mathrm{T}_1};$$

that is, the rise of temperature during compression is the same for each stage when the work of compression is a minimum; for between each stage the air is cooled down to the atmospheric temperature T_1.

When the air at the pressure p_3 and temperature T_1 is used to drive a motor, working in two stages, the work available will be given by

$$\mathrm{U}'=\frac{\mathrm{CT}_1\cdot n}{n-1}\left\{2-\left(\frac{p_2}{p_3}\right)^{\frac{n-1}{n}}-\left(\frac{p_1}{p_2}\right)^{\frac{n-1}{n}}\right\}.$$

3. *Multiple-stage Compression and Expansion.*—If the compression and expansion be carried out in three or more stages, the same conditions hold for least work as for the two-stage process; that is to say, we must have

$$\frac{p_2}{p_1}=\frac{p_3}{p_2}=\frac{p_4}{p_3}=\ldots=\frac{p_n}{p_{n-1}},$$

and the rise of temperature during each stage will be the same.

Hence, knowing the initial and final pressures (p_1 and p_n) desired, the compression pressures for the intermediate stages can readily be obtained from the above equations, and thus the actual volumes of the cylinders themselves.

§ 28. **Loss of Work due to Air Mains.**—To cause the flow of air in the pipes connecting the air-compressor or reservoir to the motor, there must be a drop of pressure in the pipes. Let (p_4-p) in fig. 29 be this fall of pressure. Then considering a single-stage compressor and motor, 2A becomes the curve of compression, and *ab* the expansion curve for the motor instead of 3B. Hence the shaded area represents work gained in the motor; but the work represented by *cd*34 is lost, and this work lost will be found to be greater than the amount gained. The actual values of these areas can be readily calculated by the formulæ already given, and may be left as an exercise to the reader. Evidently

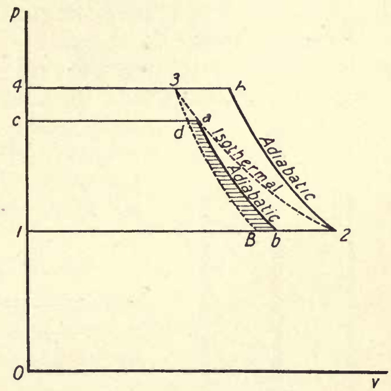

FIG. 29.—Loss of Work in Pipes.

$$\eta_{\text{motor}}=\frac{\text{area } cab1}{\text{area } ca21}; \qquad \eta_{\text{compressor}}=\frac{\text{area } 4321}{\text{area } 4A21};$$

$$\eta_{\text{mains}}=\frac{ca21}{4321}.$$

$$\therefore \quad \eta_{\text{plant}}=\eta_{\text{compressor}}\times\eta_{\text{mains}}\times\eta_{\text{motor}}=\frac{\text{area } cab1}{\text{area } 4A21}.$$

It should be noted from the above that frictional losses due to the flow of air through pipes are not dead losses, as is the case with water transmission, as the friction raises the temperature of the air and increases its internal energy. The velocity of air transmission can be from 30 to 50 feet a second as compared with 6 feet a second for water, for the same loss by friction.

CHAPTER VI.

THE RECIPROCATING STEAM-ENGINE.

§ 29. **Calculation of Heat received and Work done by the Steam in an Engine Cylinder.**—Suppose an engine cylinder contains a volume S of water upon which rests a piston weighing p lbs. per unit area. Let heat be supplied to the cylinder until all the water is evaporated. As steam is formed the piston will rise until the steam finally occupies some volume, say V. The external work done by the steam will be measured by the product

$$p \times \text{area of the piston} \times (V - S).$$

If v and s represent the volumes per lb. of steam and water under these conditions, the work done per lb. of steam will be given by

$$p(v-s)=pu \text{ (say)}.$$

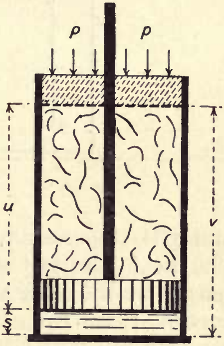

FIG. 30.

The total heat supplied to do this work will be the amount of heat necessary to raise 1 lb. of the water to the evaporation temperature corresponding to the pressure p, plus the amount of heat required to evaporate it at this temperature. Calling these quantities H, h, and L respectively, we get

$$H=h+L;$$

or if each lb. of water has only a fraction q evaporated,

$$H=h+qL.$$

For every pressure there is a corresponding temperature when the steam is just dry and saturated, and it is found that for any temperature t on the Fahrenheit scale

$$H = 1085 + \cdot 305t$$

and

$$L = 1114 - \cdot 695t$$

for dry steam.

We can represent these results very clearly by the graphical method given in Chapter I.

Referring to fig. 31, let A be the state of the water in the cylinder initially. Then AC will represent the rise of temperature and pressure of the water as heat is supplied,

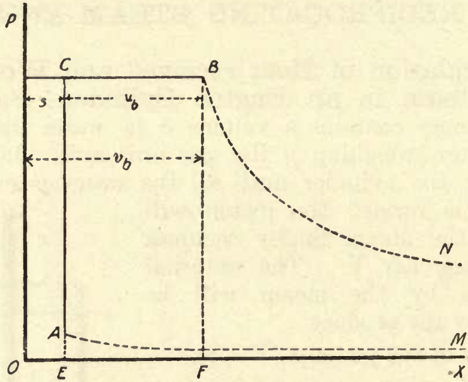

FIG. 31.

and CB the evaporation of the water at constant pressure. The heat received during the operations ACB will be represented by the area MACBN, where BN and AM are adiabatic curves drawn from B and A (see § 4), and the external work done by the area CBFE.

$$\therefore \text{ Area MACBN} = h + q_b L\ ;$$

$$\text{area CBFE} = p_b q_b \frac{(v - s)}{J}\ ;$$

and the change of intrinsic energy from the state A to the state B is given by

$$I_B - I_A = \text{area NBFX} - \text{area MAEX}$$

$$= h_b + q_b L_b - p_b q_b \frac{(v - s)}{J},$$

where J represents "Joule's equivalent."

Hence to find the change of intrinsic energy between two states, calculate the intrinsic energy in each state from water at 32° F., and subtract.

In actual engines it is not convenient in practice to evaporate water in the cylinders themselves, so evaporation is performed in a separate boiler, and the steam led to the cylinders after formation. The heat supply will be identical, neglecting losses, wherever the steam is formed, so that the observation of this practical necessity will not alter the heat account.

Consider, therefore, the indicator diagram obtainable from an engine cylinder when steam is thus supplied, as shown in fig. 32.

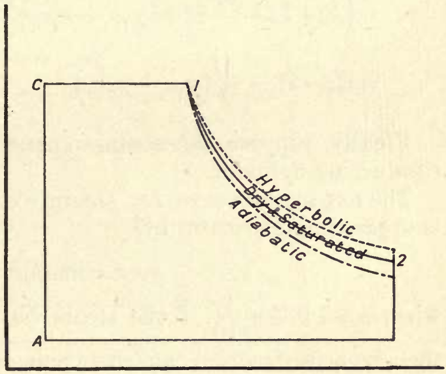

FIG. 32.

For economy, the steam supply is usually cut off after a portion of the piston stroke, as shown at 1; and then the steam expands owing to its own elasticity, for the remainder of the stroke, as shown at 12. The heat received during AC1 we have just seen how to calculate, and are now only concerned with the quantity of heat supplied during 12.

Suppose the expansion 12 be *hyperbolic.* Then at any point of the expansion

$$pqv = \text{constant}.$$

Now, Heat received during 12
= Intrinsic energy of the steam at 2
+ Work done between the states 1 and 2
− Intrinsic energy of the steam at 1

$$=\left(h_2+q_2\mathrm{L}_2-\frac{p_2q_2v_2}{\mathrm{J}}\right)+p_1q_1v_1\log_e r-\left(h_1+q_1\mathrm{L}_1-\frac{p_1q_1v_1}{\mathrm{J}}\right),$$

neglecting s, which is small.

This measures the heat that must be given to the steam during expansion by the cylinder jackets.

Next suppose the expansion curve 12 is a *dry and saturated* steam curve.

For dry steam it is found that

$$pv^{\frac{17}{16}} = 475,$$

when p is measured in lbs. per sq. in. and v in cub. ft., gives approximately the curve of expansion. Hence the heat received during 12

$$= \left(h_2 + L_2 - \frac{p_2 v_2}{J}\right) + \left(p_1 v_1 - p_2 v_2\right)\frac{16}{J} - \left(h_1 + L_1 - \frac{p_1 v_1}{J}\right);$$

$$= H_2 - H_1 + \frac{17}{J}\left(p_1 v_1 - p_2 v_2\right).$$

Finally, suppose the steam expand *adiabatically* in a non-conducting cylinder.

The expansion curve for steam expanding in this manner is approximately given by

$$pv^n = \text{constant}$$

when $n = 1{\cdot}035 + \frac{q_1}{10}$, if the steam be not too wet, and q_1 is the dryness fraction at the commencement of expansion. Hence now

$$\text{Heat received during } 12 = 0$$

$$= \left(h_2 + q_2 L_2 - \frac{p_2 q_2 v_2}{J}\right) + \frac{p_1 q_1 v_1 - p_2 q_2 v_2}{n-1} - \left(h_1 + q_1 L_1 - \frac{p_1 q_1 v_1}{J}\right);$$

from which we can find q_2 the dryness fraction at the end of expansion, when q_1 is known.

If the three curves of expansion be plotted on a volume base, they will be as shown in fig. 32, when the pressure and volume at the commencement of expansion is the same in each case. It should be observed that since the volume is least for any given pressure when the expansion is *adiabatic* and $n = 1{\cdot}35$ (when $q_1 = 1$), that is when no heat is supplied during the expansion, the steam must get wetter as it expands than when the expansion is *dry and saturated* and $n = 1{\cdot}06$. Also, when the expansion is *hyperbolic* and $n = 1$, the steam will get drier and may be superheated even

at the end of expansion. Hence both for dry and saturated, and for hyperbolic expansion, heat must be given to the cylinder by some form of steam jacket during the expansion.

§ 30. **Calculation of Work done from Actual Indicator Diagrams.**—Let the shaded area of fig. 33 be an indicator diagram taken from the cylinder of a direct-acting steam-engine. The pressure per unit area driving the piston at any point of the stroke is given by the ordinate of the diagram at the point considered. For calculation purposes it is necessary to find the mean pressure which, acting throughout the stroke, will give the same work as that

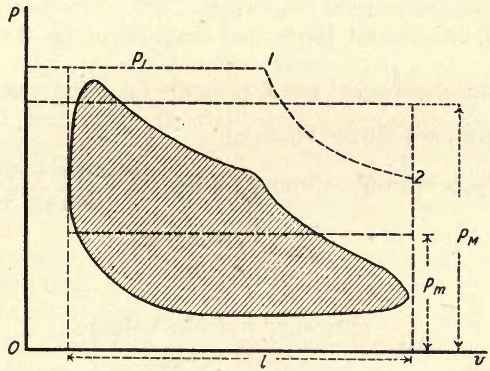

FIG. 33.

actually performed. This mean pressure (p_m) can be found by measuring the area of the indicator diagram by a planimeter and dividing by the length of the diagram, or else by drawing a number of ordinates at equal distances apart on the diagram and finding the mean length of these ordinates.

The work done per stroke of the piston will then be simply ($p_m A \times l$), where l is the stroke and A the area of the piston.

If the engine be double-acting, the work per revolution will be $2p_m A . l$.

Hence the indicated horse-power of such an engine will be given by the formula

$$\text{I.H.P.} = \frac{2p_m . A . l \times N}{33000},$$

where N is the number of revolutions per minute, l is the length of stroke in feet, and $p_m A$ is the load on the piston in lbs.; since one horse-power is the rate of doing 33,000 foot-lbs. of work in one minute.

In designing an engine cylinder to produce a given horse-power, with a given initial steam pressure p_1 at admission, a given point of steam cut-off in the cylinder, and a known number of revolutions and length of stroke, we have to find the area of the cylinder. To do this, the mean pressure necessary throughout the stroke must be obtained. p is not known *ab initio*, but it can be found as follows:—

Draw the theoretical diagram for the given steam pressure p_1, cut-off, etc., round the actual diagram of fig. 33, as shown dotted.

Then the theoretical mean pressure (p_M) is given by

$$p_M \times (\text{clearance} + \text{stroke volume})$$

$$= p_1 \times \text{cut-off volume} \left(1 + \log_e \frac{\text{clearance} + \text{stroke vol.}}{\text{cut-off vol.}}\right);$$

or

$$p_M = p_1 \frac{(1 + \log_e R)}{R}$$

where

$$R = \frac{\text{clearance} + \text{stroke volume}}{\text{cut-off volume}},$$

and the expansion during 12 is assumed to be hyperbolic. It is found by experience that for all engines of a similar type, the ratio

$$p_m : p$$

is practically a constant. Hence knowing p_1, etc., we can calculate p_M, and so obtain p_m and thence the cylinder area required by substituting the known data in the formula for the indicated horse-power. For single-cylinder condensing engines with a ratio of expansion given by $R =$ about 1·6,

$$\frac{p_m}{p_M} = \cdot 45.$$

This gives approximately the true mean pressure obtained during the stroke. It should be noted that p_m is the mean pressure during the stroke volume only, while p_M is the mean pressure for the stroke volume plus the clearance volume.

§ 31. **Combined Indicator Diagrams.** — In large engines it is more economical, for practical reasons, to expand the steam in stages in successive cylinders, than to expand it completely in one cylinder. The range of temperature in each cylinder between the temperatures of the admission and exhaust steam is thus less than if the whole expansion were carried out in one cylinder, and so less loss by condensation takes place. For least loss the total temperature range in each cylinder should be the same. The total work done is the same whether the expansion is carried out in stages in a number of separate cylinders or all at once in one cylinder, so long as the total ratio of expansion is the same. Thus the combined diagrams of a three-cylinder triple expansion engine would be as shown in fig. 34; and for a given initial pressure OA and cut-off volume AB, the theoretical work area OABDL will be unaltered no matter how many stages it is performed in, so long as the ratio of the volume OL to the volume AB is unaltered. The *actual* indicator diagrams from the high-pressure, intermediate, and low-pressure cylinders, when drawn to the same scales of pressure volume, will be somewhat as shown shaded in the figure, where OC_H is the clearance volume and C_HH is the stroke volume of the high-pressure cylinder; and OC_I and C_II, and OC_L and C_LL, the corresponding volumes for the intermediate and low-pressure cylinders.

The mean pressure p_m which, acting throughout the stroke volume of the low-pressure cylinder (*i.e.* along C_LL), would give the same work as is represented by the three actual indicator diagrams, can easily be found by measuring their area with a planimeter.

The theoretical mean pressure p_M, which will be represented by the mean height (PQ) of the area OABDL, is given by

$$p_M = p_1 \frac{(1+\log_e R)}{R},$$

where
$$R = \frac{OL}{AB} = \frac{OC_L + C_LL}{AC+CB} = \frac{r\,(1+C_L)}{m+C_H};$$

and
$$r = \frac{\text{volume of L.P. cylinder}}{\text{volume of H.P. cylinder}};$$

$$m = \frac{\text{cut-off volume of H.P. cylinder}}{\text{stroke volume of H.P. cylinder}};$$

$$C_H = \frac{\text{clearance volume of H.P. cylinder}}{\text{stroke volume of H.P. cylinder}};$$

$$C_L = \frac{\text{clearance volume of L.P. cylinder}}{\text{stroke volume of L.P. cylinder}}.$$

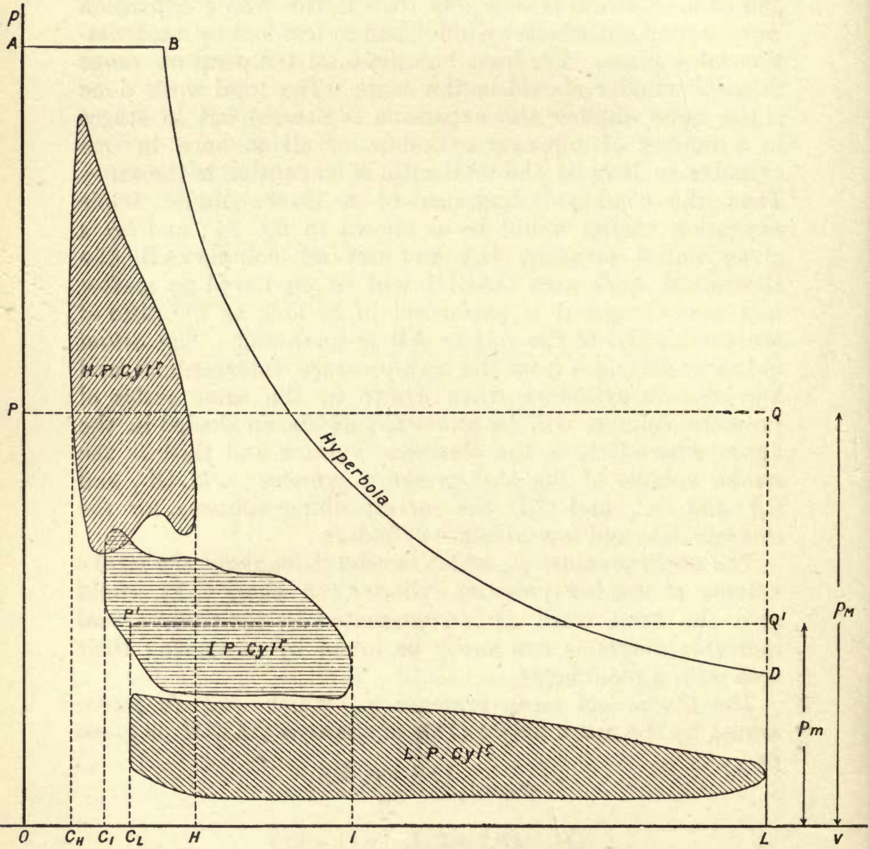

FIG. 34.—Combined Indicator Diagrams from a Triple Expansion Engine. Scales: 1 cm. = 2 cub. ft. volume = 10 lbs. pressure per sq. in.

In modern triple expansion engines when p_1 is from 200 to 250 lbs. per sq. in., r is usually about 7. It is then found that by taking C_H and C_L equal to 25 per cent. and 15 per

cent. respectively, that for a given cut-off (m), not outside the limits of 65 per cent. to 75 per cent. of the stroke, the ratio of p_m to p_M as found above is practically constant, and is given by

$$\frac{p_m}{p_M} = 0{\cdot}49.$$

The factor 0·49, known as the *design factor*, is thus greater for a three-cylinder than for a single-cylinder engine, which is due to the greater economy of the former type.

In engine design we start with p_1 and R as the known quantities and so obtain p_M, and then take p_m as $\cdot 49 P_M$. Having thus found p_m we can at once get the area, and hence the volume, of the low-pressure cylinder required for a given power by substituting for p_m in the indicated horse-power formula. The volume of the high-pressure cylinder will then be $\frac{1}{r}$, say $\frac{1}{7}$th, of this, and the volume of the intermediate cylinder is then about $\frac{1}{\sqrt{r}}$, *i.e.* $\frac{1}{\sqrt{7}}$, say, of the low-pressure cylinder volume, for this ratio of cylinder volumes has been found to give in practice about an equal range of temperature in each cylinder.

§ 32. **Calculation from Actual Indicator Diagrams of the Transfer of Heat between the Steam and Cylinder Walls. Hirn's Analysis.**—Hirn's analysis gives a method by which we can calculate the quantity of heat given to or taken from the cylinder walls by the steam during any portion of the cycle of operations through which it passes.

Suppose we have an actual indicator diagram as *abcd*. Let 0123 be the points of admission, cut-off, release, and compression respectively, and let a represent the operation from 0 to 1, b that from 1 to 2, and so on.

Then during any operation it follows:—

$$\left.\begin{array}{l}\text{The heat received from}\\ \quad\text{external sources}\end{array}\right\} = \left\{\begin{array}{l}\text{The work done by the steam}\\ +\text{the increase of its internal}\\ \qquad\text{energy}\\ +\text{any heat given to the}\\ \qquad\text{cylinder walls.}\end{array}\right.$$

Let h = the sensible heat of water from 32° F.;
L = latent heat at any temperature;
ρ = the internal energy of evaporation $= L - \frac{p(v-s)}{J}$;
q = the dryness fraction at any point;
w_c = the weight of the cylinder cushioning steam;
w_f = the weight of feed water used per stroke;
U = the work done by the steam during any operation;
Q = the heat given to the cylinder walls during the same operation.

Then if symbols without suffices refer to boiler steam, and those with suffices to the points 0, 1, 2, 3 on the diagram, we

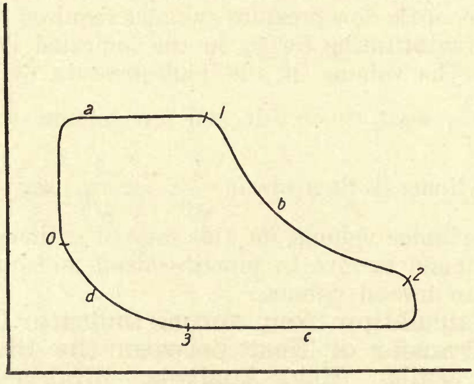

FIG. 35.

shall, on the assumption that the expansion during b is adiabatic, obtain the following results:—

From 0 *to* 1:

Heat received from 32° F. $= w_f(h+qL) = U_a + (w_f+w_c)(h_1+q_1\rho_1) - w_c(h_0+q_0\rho_0) + Q_a$;

From 1 *to* 2:

* Heat received $= 0 = U_b + (w_f+w_c)(h_2+q_2\rho_2) - (w_f+w_c)(h_1+q_1\rho_1) + Q_b$;

From 2 *to* 3:

Heat received $= -c(t_f-t_i) = -U_c + w_c(h_3+q_3\rho_3) + w_f h_4 - (w_f+w_c)(h_2+q_2\rho_2) + Q_c$;

* *N.B.*—For a jacketed engine we must write Q_J for 0 on the left-hand side of the equation.

where c represents the lbs. of condenser circulating water per stroke; t_i and t_f the initial and final temperatures of the circulating water as it passes through the condenser; and t_4 the temperature of the condensed steam.

From 3 *to* 0:

$$\text{Heat received} = 0 = -\mathrm{U}_d + w_c(h_0 + q_0\rho_0) - w_c(h_3 + q_3\rho_3) + \mathrm{Q}_d.$$

We write $-\mathrm{U}_c$ and $-\mathrm{U}_d$, since this work is not done *by* the steam, but *on* it.

Summing up all the heat received, we get—

$$w_f(h - h_4 + q\mathrm{L}) - c(t_f - t_i) = \mathrm{U}_a + \mathrm{U}_b - \mathrm{U}_c - \mathrm{U}_d + \mathrm{Q}_a + \mathrm{Q}_b + \mathrm{Q}_c + \mathrm{Q}_d.$$

In the actual problem, w_f, c, t_f, t_i, and t_4 can be directly measured by suitable apparatus. From the indicator diagram, U_a, U_b, U_c, and U_d—that is, the external work done by the steam during any portion of the piston stroke—can be obtained. h, ρ, and L are obtained from steam tables when the pressure of the boiler steam is known; q can be measured by a suitable dryness fraction calorimeter.

Hence to obtain Q_a, Q_b, etc., the only remaining quantities to be determined are w_c, q_0, q_1, q_2, and q_3 Let u be the volume of the cylinder at any point, and let v be the specific volume of 1 lb. of steam at any given pressure. Then—

$$w_c q_0 v_0 = u_0\,; \quad (w_f + w_c) q_1 v_1 = u_1\,;$$
$$w_c q_3 v_3 = u_3\,; \quad (w_f + w_c) q_2 v_2 = u_2\,;$$

for at the points 0 and 3 there is only the cushion weight of steam in the cylinder.

There are thus four equations from which to find five unknowns. Hence an assumption must be made. If the clearance and amount of cushioning in the cylinder be small, we can assume $w_c = 0$; but usually a truer assumption to make will be to write $q_3 = 1$—that is, assume the steam dry at the point of compression.

We now have a sufficient number of equations, and hence can find Q_a, Q_b, Q_c, and Q_d from the original equations given. Evidently if the cylinder be unjacketed the sum $\mathrm{Q}_a + \mathrm{Q}_b + \mathrm{Q}_c + \mathrm{Q}_d$ must be positive, as it must represent the loss by conduction and radiation from the cylinder walls. If the

cylinder be jacketed, we must write Q_J for 0 in the second equation, where Q_J is measured by the amount of water drained from the jacket multiplied by the latent heat of the steam at the temperature of condensation in the jacket.

§ 33. **Indicated Steam Consumption in an Engine Cylinder.**—The weight of steam used per stroke in an engine cylinder can be calculated from the engine indicator card, on the assumption that the steam at the points of cut-off and compression is *dry and saturated.* Consider

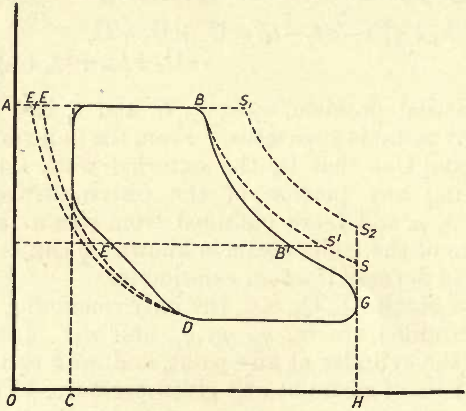

FIG. 36.

the indicator card shown in fig. 36. Let OC represent the clearance volume of the cylinder and CH the stroke volume. Let AB be the volume of steam in the cylinder at the point of cut-off, and through B draw a dry and saturated steam curve BS for the total steam in the cylinder. To take account of the clearance steam which is shut into the cylinder during compression, and is therefore present at admission on the next stroke, draw a dry and saturated steam curve DE through the point of compression D.

AB is then the volume of total dry steam in the cylinder at cut-off, and AE is the volume of dry clearance steam in the cylinder at this point. Hence EB is the volume of the working steam which passes through the cylinder per stroke.

Hence

$$\left.\begin{array}{l}\text{The indicated weight of}\\ \quad\text{steam used per stroke}\\ \quad\text{in lbs.}\end{array}\right\} = \left\{\frac{\text{Volume of steam EB}}{\begin{array}{c}\text{Volume of 1 lb. of dry steam}\\ \text{at the pressure EB.}\end{array}}\right.$$

If the actual expansion curve of the indicator diagram be as BG, we can, by measuring the ratio of several such points as E′B′ to E′S′, tell whether the steam in the cylinder is drying or the reverse during expansion.

If the steam be wet at cut-off, we cannot get the true consumption of steam per stroke; but if we have some means of measuring the actual weight of feed water used per stroke we can plot a dry steam curve S_1S_2 for this actual weight, and then the ratio of the volume EB of working mixture in the cylinder to the volume ES of measured dry feed steam at the same pressure, will be the *dryness fraction* of the cylinder steam at that point. All this assumes the clearance steam dry throughout the expansion, for we have measured our volumes from the dry steam curve DE. It would probably be more correct to assume that the steam in the clearance space was in the same state as that in the cylinder at every point during expansion. The point E would then have to be set back to E_1, so that

$$AE_1 : AE = EB : ES_1,$$

and then the more accurate dryness fraction at B would be given by $\frac{E_1B}{ES_1}$.

It should be noted that when the diagrams from several cylinders are combined, as in fig. 34, we cannot draw one single continuous dry saturation steam curve for all the cylinders, as the *total steam* expanding in each cylinder is different owing to the clearance volumes of each cylinder being different.

Hence a separate saturation curve must be drawn for each diagram, unless the diagrams be "set back" and the effect of clearance eliminated, in which case one continuous dry steam curve can be drawn for the *working steam* which passes through all the cylinders in turn, and so is therefore of constant quantity.

This "setting back" is effected by drawing compression

curves, such as DE in fig. 36, through the point of compression of each diagram in fig. 34, and then setting back all such points as E and E′ (fig. 36) until they coincide with the zero volume line OA, and every point on the expansion curves, as BB′, the same amount at the corresponding pressure. Thus new curves of expansion are obtained for each cylinder, and now the volume of steam measured from the zero line OA to these curves at any pressure will give the volume of the "working steam" passing through all the cylinders in turn at that pressure.

CHAPTER VII.

UNRESISTED EXPANSION, AND FLOW THROUGH ORIFICES.

§ 34. **Effect of Unresisted Expansion.**—So far we have assumed that when any gas has passed through any cycle of operations, its expansive force has at every point been exactly balanced by external resistances.

If a substance be allowed to expand without overcoming resistance and doing external work, the energy of expansion becomes available for other purposes. It may simply appear as kinetic energy in the substance itself, as, for instance, when steam escapes from a boiler through an orifice into the air. Here energy appears first as a violent agitation of the particles of the substance, and ultimately, as the motion subsides, the kinetic energy of the motion reappears in the form of heat. The available energy may also be utilised to increase the velocity of flow of the substance, as in the case of gases flowing through pipes, or yet again to overcome frictional resistances of pipes, etc.

If the expansive energy of the substance were developed in some form of motor cylinder, we could, by the methods already discussed, calculate the work that could be done on external objects by the substance when the conditions under which the operations took place and the initial and final states of the substance were known.

If no motor be used, then this same amount of work will generate kinetic energy in the substance, which, if not used to cause flow or overcome frictional resistances, will finally reappear as heat in the substance. (The terminal state would not be reached until temperature equilibrium had

been reached.) Hence, using a motor, the substance will be in a certain state at the final pressure; while when no motor is used, the substance will be in a *higher* state at the same terminal pressure. This is true of all substances, whether heat is received or not, during their passage through the expansion nozzle. It is, in fact, simply the statement of the First Law of thermo-dynamics on the assumption that there is no energy of motion either before or after the unresisted expansion.

We must therefore find out how much work the substance could do if it expanded in a motor under the given conditions, and also find the final state of the substance under these conditions. We can then deduce its final state when the expansion is unresisted.

§ 35. **Increase of Thermal State due to Unbalanced Expansion.**—The two most common cases of unbalanced expansion are:—

(1) When steam or gas escapes from a region of constant pressure to another region of lower constant pressure, as in wire-drawing at the throttle valve of a steam-engine, or at the expansion valve of a refrigerator.

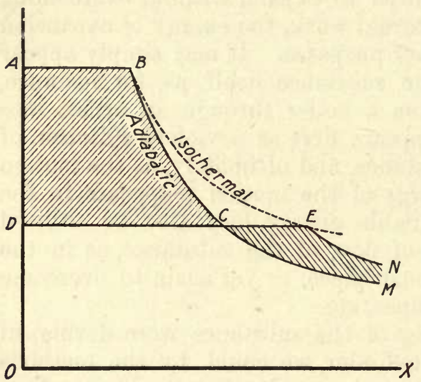

Fig. 37.

(2) When steam or gas is discharged from a vessel of constant volume into a region of constant pressure, as when steam exhausts from an engine cylinder into a receiver.

The difference between the two cases is, that in the first the substance is continuously supplied at the higher pressure, while in the second case there is only a given volume of the substance to expand. Consider case (1). Imagine the substance to flow along a pipe at pressure p_1, represented by OA in fig. 37, and to leave it at pressure p_2, given by OD. Assume the reduction in pressure takes place at the valve without receiving or rejecting heat, *i.e.* adiabatically.

Suppose the reduction in pressure had been effected in a motor cylinder without clearance; then the work done in the motor would have been the area ABCD. At the lower pressure p_2, the point C in the diagram gives the final state of the substance under these conditions. Now if the expansion be unresisted, the state will not be C but E where the area MCEN (representing the heat received during CE) is equal to the area ABCD.

If the unbalanced expansion be not adiabatic, then the state will not be E, but will be a higher or lower state, though always at the pressure p_2, according as heat is received or rejected at the expansion valve.

Apply these principles to the unresisted expansion of a perfect gas.

For a perfect gas

$$pV = CT \text{ and } pV^{\gamma} = \text{constant},$$

where $\gamma = \frac{K_p}{K_v}$ when the expansion is adiabatic.

If the ratio of adiabatic expansion R is given by $\frac{V_c}{V_B}$, we know $\frac{T_c}{T_B} = \frac{1}{R^{\gamma - 1}}$, and thus the temperature at the pressure OD, if a motor were used, is at once found.

The temperature T_E, which the gas actually acquires when the expansion is unresisted, is given by

$$K_p(T_E - T_c) = \text{area ABCD};$$

$$= (p_B V_B - p_c V_c)\frac{\gamma}{\gamma - 1};$$

$$= (T_B - T_c)\frac{CK_p}{K_p - K_v};$$

$$= K\ (T_B - T_c).$$

Hence $$T_E = T_B.$$

Therefore when a *perfect gas* is wire-drawn from one pressure to a lower one without gain of kinetic energy, the temperature is unaltered after expansion.

Next consider case (2), where only a certain mass of fluid is allowed freely to expand. Let its initial state be repre-

sented by B in fig. 38; then if the substance were expanded in a motor the nett work it could do would be represented by the area BGC, which is obviously much less than in case (1). Hence if expansion be unresisted, the final state E of the substance is obtained by making the area MCEN equal to BGC, where EN and CM are adiabatic curves drawn to infinity through E and C.

If the expansion from the vessel of constant volume be carried to its limit, E will not only denote the state of the substance which has escaped but also of that in the receiver itself.

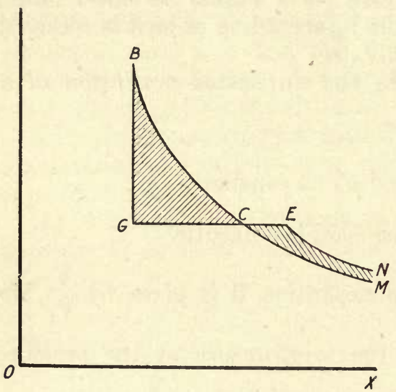

Fig. 38.

§ 36. **Discharge through Orifices.**—As has just been pointed out, in all cases where unresisted expansion takes place from a region of constant pressure to another of lower pressure, the work area ABCD of fig. 37 is available to (*a*) cause increase of kinetic energy in the substance; (*b*) do work against gravity; (*c*) do work in overcoming frictional resistances.

In considering the discharge through orifices we can neglect (*b*) and (*c*) in the case of gases or steam, as they are very small compared to (*a*), though in the case of such fluids as water or oil this would not be justifiable.

Approximately, therefore, for gases and steam we may write

$$\text{Area ABCD of fig. 37} = \text{Increase of kinetic energy}$$

$$= \frac{\sigma_2^2 - \sigma_1^2}{2g} \text{ per lb.}$$

where σ_1 and σ_2 are the velocities of the gas where the pressures are p_1 and p_2. Usually σ_1 is very small compared to σ_2, so we can write

$$\text{Area ABCD} = \frac{\sigma_2^2}{2g}.$$

This velocity σ_2 in general will eventually subside at a certain distance from the orifice of discharge, and the kinetic energy $\left(\frac{\sigma_2^2}{2g}\right)$ will again be converted into heat, unless the discharge be used to produce flow in pipes, etc. In order to find σ_2 we must know the nature of the expansion curve BC.

§ 37. **Hyperbolic Discharge through Orifices.**—Suppose that the curve of expansion BC is given by

$$p \times u = \text{constant},$$

which means that the expansion in the discharging orifice is isothermal in the case of a perfect gas, and hyperbolic in the case of steam or other vapours.

We then have

$$\text{Area ABCD} = \frac{\sigma_2^2}{2g} = \int_2^1 u dp = p_1 u_1 \log_e \frac{p_1}{p_2} \quad . \quad (1)$$

If (a) be the contracted area of the issuing jet at which the pressure is uniform and equal to p_2, and if W be the discharge in lbs. per second, and u_2 the volume per lb. of the gas at the pressure p_2; then, evidently—

$$W = \frac{a.\sigma_2}{u_2} = \frac{ap_2}{p_1 u_1} \sqrt{2gp_1 u_1 \log_e \frac{p_1}{p_2}} \text{ from } \quad (1)$$

$$= \frac{ap_2 \sqrt{2g}}{\sqrt{p_1 u_1}} \sqrt{\log_e \frac{p_1}{p_2}} \quad . \quad . \quad . \quad (2)$$

For a given initial pressure p_1 and volume u_1 per lb., this expression is a maximum when

$$\sqrt{\log_e \frac{p_1}{p_2}} - \frac{p_2}{2\sqrt{\log_e \frac{p_1}{p_2}}} \times \frac{1}{p_2} = 0,$$

or

$$\log_e \frac{p_1}{p_2} = \tfrac{1}{2}; \text{ or } \frac{p_2}{p_1} = \cdot 606,$$

as found by differentiating expression (2).

Substituting for $\frac{p_2}{p_1}$, we then get the *maximum discharge* given by

$$W_{\text{max.}} = \frac{a\sqrt{2g}}{\sqrt{p_1 u_1}} \times \frac{\cdot 606 p_1}{\sqrt{2}} = \frac{3 \cdot 44 p_1 a}{\sqrt{p_1 u_1}}.$$

That there must be a maximum theoretical discharge for some particular value of $\frac{p_2}{p_1}$ is obvious when we consider that, though σ_2 will increase with the drop in pressure, yet $\frac{1}{u_2}$, that is, the weight per cub. ft. of the gas, will decrease as p_2 decreases, which will limit the flow.

Let us apply the above results to some particular cases. Take *air*, for instance, where for hyperbolic flow

$$p_1 u_1 = 53{\cdot}2 T_1 ;$$

T_1 being measured on the absolute temperature scale.

Then

$$W_{\text{max.}} = \frac{{\cdot}47 p_1 . a}{\sqrt{T_1}} \text{ lbs. per second;}$$

and the corresponding velocity of discharge, σ_2, is given by

$$\sigma_2 = \sqrt{2 g p_1 u_1 \log_e \frac{p_1}{p_2}} = \sqrt{\frac{2g \times 53{\cdot}2T_1}{2}} = 41{\cdot}4 \sqrt{T_1} \text{ ft. per sec.}$$

Or, again, take the case of *dry steam*, where for hyperbolic expansion

$$p_1 u_1 = 253^2 \text{ per lb. of steam.}$$

Then

$$W_{\text{max.}} = \frac{3{\cdot}44 p_1 a}{253} = \frac{p_1 a}{73{\cdot}5} ;$$

and

$$\sigma_2 = \sqrt{\frac{2g \times 253^2}{2}} = 1435 \text{ ft. per second.}$$

If a be measured in square inches, p_1 must be taken as the pressure per square inch; and if a is in square feet, then p_1 must be the pressure per square foot.

§ 38. **Discharge of Gases through Orifices with any Type of Expansion.**—Suppose, now, instead of hyperbolic discharge, the gas expands as it passes through the orifice according to the law

$$pu^n = \text{constant.}$$

$\therefore$ Area ABCD of fig. 37

$$=\frac{n}{n-1}(p_1u_1-p_2u_2);$$

$$=\frac{n}{n-1}\cdot p_1u_1\left\{1-\left(\frac{p_1}{p_2}\right)^{\frac{1-n}{n}}\right\};$$

$$=\frac{\sigma_2{}^2}{2g}.$$

As before, the discharge for given limits of pressure p_1 and p_2 is given by

$$\mathrm{W}=\frac{a\sigma_2}{u_2}=a\sqrt{\frac{\frac{n}{n-1}2gp_1u_1\left\{1-\left(\frac{p_1}{p_2}\right)^{\frac{1-n}{n}}\right\}}{u_1\left(\frac{p_1}{p_2}\right)^{\frac{1}{n}}}};$$

$$=\frac{a}{u_1}\sqrt{\frac{n}{n-1}2gp_1u_1}\left\{\left(\frac{p_2}{p_1}\right)^{\frac{2}{n}}-\left(\frac{p_2}{p_1}\right)^{\frac{n+1}{n}}\right\}^{\frac{1}{2}}.$$

So long as p_2 is not zero, this expression is a maximum for a given p_1 when

$$\frac{2}{n}\left(\frac{p_2}{p_1}\right)^{\frac{2}{n}-1}-\frac{n+1}{n}\left(\frac{p_2}{p_1}\right)^{\frac{1}{n}}=0;$$

or

$$\frac{p_2}{p_1}=\left(\frac{2}{n+1}\right)^{\frac{n}{n-1}}.$$

Substituting this value in the expression for W, we get

$$\mathrm{W}_{\max.}=\frac{a\sqrt{2g}\cdot p_1}{\sqrt{p_1u_1}}\left(\frac{n}{n+1}\right)^{\frac{1}{2}}\left(\frac{2}{n+1}\right)^{\frac{1}{n-1}};$$

and the corresponding velocity of discharge given by

$$\sigma_2=\sqrt{2gp_1u_1\cdot\frac{n}{n+1}}.$$

Thus for *air* expanding adiabatically $n=\gamma=1{\cdot}408$—

$$\therefore\ \frac{p_2}{p_1}=\cdot 528;$$

$$\therefore\ \mathrm{W}_{\max.}=\frac{\cdot 534p_1a}{\sqrt{\mathrm{T}_1}}\text{ lbs. per second,}$$

with a velocity

$$\sigma_2=44{\cdot}7\sqrt{\mathrm{T}_1}\text{ ft. per second.}$$

Or, again, for the adiabatic expansion of *steam*, $n=1{\cdot}13$—

$$\therefore\ W_{\text{max.}}=\frac{p_1a}{70},\ \text{with a velocity of discharge}$$

given by $$\sigma_2=5{\cdot}84\sqrt{p_1u_1}.$$

For *superheated steam*, $n=1{\cdot}3$ approximately for adiabatic expansion. Therefore

$$W_{\text{max.}}=\frac{3{\cdot}78p_1a}{\sqrt{p_1u_1}},$$

and $$\sigma_2=6{\cdot}03\sqrt{p_1u_1}.$$

We can also calculate the discharge for saturated steam of different degrees of moisture; for if v_1 be the volume of 1 lb. of *dry* steam at the pressure p_1 and q_1, the dryness fraction, then $$qv_1=u_1.$$

Hence substituting when

$$\begin{array}{rllll} q_1= & 1 & \cdot75 & \cdot5 & \cdot25; \\ \dfrac{\sqrt{p_1v_1}}{p_1a}\times W_{\text{max.}}= & 3{\cdot}62 & 4{\cdot}12 & 5{\cdot}0 & 6{\cdot}83; \\ \dfrac{\sigma_2}{\sqrt{p_1v_1}}= & 5{\cdot}84 & 5{\cdot}04 & 4{\cdot}07 & 2{\cdot}08. \end{array}$$

§ 39. **Discharge through Orifices due to Small Pressure Differences.**—The foregoing results have been obtained on the assumption that the difference of pressure between p_1 and p_2 on the two sides of the orifice was finite and considerable. Now, suppose this pressure difference to be quite small. Then we may write

$$\frac{\sigma_2^{\ 2}}{2g}=u_{\text{mean}}\times\Delta p=u_1\Delta p;$$

for since the pressure difference Δp is small, u_{mean} will not appreciably differ from u_1 or u_2.

$\therefore$ In this case

$$W=\frac{a\sigma_2}{u_2}=\frac{a\sqrt{2g}}{\sqrt{p_1u_1}}\sqrt{p_1\cdot\Delta p}\ \text{lbs. per second.}$$

Thus for *air*,

$$W=1{\cdot}1a\sqrt{\frac{p_1\Delta p}{T_1}},$$

a formula applicable to air escaping from a stokehold under air pressure.

For *dry steam*, $$W = \frac{a}{31{\cdot}6}\sqrt{p_1 \,.\, \Delta p};$$

or for *steam, of dryness q*, when

$$q_1 = 1{\cdot}0 \qquad {\cdot}75 \qquad {\cdot}5 \qquad {\cdot}25$$

$$\frac{W}{a\sqrt{p_1 \Delta p}} = \frac{1}{31{\cdot}6} \quad \frac{1}{27{\cdot}4} \quad \frac{1}{22{\cdot}3} \quad \frac{1}{15{\cdot}8}.$$

§ 40. **Experimental Work on the Discharge of Gases through Orifices.**—In the foregoing theoretical considerations we have assumed that the velocity σ_2 and the pressure p_2 are constant all over the area (a) of the orifice. In other words, we have assumed that the expansion of the escaping fluid has been completed by the time the section (a) has been reached.

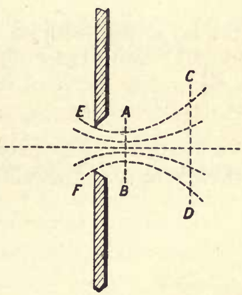

FIG. 39.

It is found by experiment that so long as the lower pressure p_2 is less than one-half the higher p_1, the discharge is practically *constant* for all values of p_2. This does not agree with the theory, so our assumption that the pressure p_2 is constant over the area (a) must be incorrect. It is found actually that though the pressure along the outer streams of a jet issuing from an orifice may be p_2, yet at the centre of the jet the pressure will be greater than p_2 at the *vena contracta* AB of fig. 39. If, however, we go further from the orifice until we reach a section of the jet over which the pressure is constant, as CD, then we can apply our theoretical results to practical cases by taking CD as the area (a) in the formula instead of the area at AB.

Napier has showed that for *dry steam*, so long as the lower pressure p_2 is equal to or less than ·6 of the higher pressure, the actual discharge is constant and equal to the maximum value calculated by theory. Thus he states that under these conditions

$$W = \frac{p_1 a}{70} \text{ lbs. per second}$$

where a is the contracted area AB in fig. 39, while for small differences of pressure

$$W = \frac{p_1 a}{34}\sqrt{\frac{\Delta p}{p_2}}.$$

For large falls of pressure it is found the contracted area AB is about ·75 the area of the orifice itself, while for small drops it is only ·66 of the actual orifice.

Other experimenters have found that for *air*, so long as p_2 is equal to or less than $\cdot 58 p_1$,

$$W = \cdot 53 \frac{p_1 a}{\sqrt{T_1}}$$

for large differences, and

$$W = 1 \cdot 1 a \sqrt{\frac{p_1 \Delta p}{T_1}}$$

for small differences of pressure. So that, on the whole, theory agrees singularly well with practice.

CHAPTER VIII.

FLOW OF GASES ALONG PIPES.

§ 41. Thermo-dynamic Difference between the Flow of Gases along Pipes and through Orifices.—In considering the flow of gases through orifices we are able to neglect frictional losses during discharge, without sensible error, and can assume that the total energy of expansion is utilised in producing velocity of the gas on the low-pressure side of the orifice.

In the case of flow along pipes frictional losses usually become the principal item, for the change in the velocity of flow along the pipes is small in practice. Hence if there is a pressure drop between two sections in a pipe, the energy of expansion due to this fall of pressure becomes dissipated by overcoming the friction between the flowing gas and the pipe, instead of in producing an increase in the velocity of the gas.

If, however, the diameter of the pipe be large in comparison to its length, the loss of energy due to friction may be very small, in which case a given drop in pressure in the pipe *will* produce an increase in the velocity of flow of the gas corresponding to the fall of pressure.

It is necessary, therefore, before writing down our energy equation for any particular case, to consider the conditions under which the flow will take place.

Thus in the case of flow through a large pipe in which the pressure drop is considerable in the pipe itself, we can equate the change of kinetic energy of the flowing gas to the work done by the pressure difference in the pipe, so long as the area of the pipe is sufficiently great to make frictional losses negligible.

A rather different case occurs when gases flow up a boiler funnel, which has a large sectional area in proportion to its length. Here the velocity of flow is practically constant at all points in the funnel, and we can also neglect any frictional losses. Hence we may consider the funnel as a large orifice, and simply equate the kinetic energy of the funnel gases to the work done by the small difference of pressure at the base of the funnel which produces the flow.

If a pipe be small and long, the velocity of flow will be practically uniform along the pipe, and in this case the work due to the drop of pressure between one end of the pipe and the other will be used in overcoming frictional resistances along the pipe; for frictional losses are not negligible when dealing with flow through small, long pipes.

§ 42. **The Flow of Gases along Large Pipes neglecting Frictional Resistance.**—As for the discharge of gases through orifices, so in all cases where there is a difference of pressure $(p_1 - p_2)$ between the two ends of a pipe through which gas is flowing, the work area ABCD of fig. 37 represents the energy available to produce flow in the pipe, to overcome gravity effects, or to overcome frictional resistances.

The effect of gravity can generally be neglected, as the density of gases is small even when they are under considerable pressure.

If, in addition, we can neglect frictional losses, our energy equation becomes

$$\frac{\sigma_2^2 - \sigma_1^2}{2g} = \int_2^1 p\,du \quad . \quad . \quad . \quad . \quad (1)$$

where σ_2 and σ_1 are the velocities of flow at the two ends of the pipe.

This equation is quite general. Thus for *water*, u is constant, and equal to $\frac{1}{w}$ where w is the weight of unit volume.

Hence for water—

$$\frac{\sigma_2^2 - \sigma_1^2}{2g} = \frac{p_1 - p_2}{w};$$

or

$$\frac{p_2}{w} + \frac{\sigma_2^2}{2g} = \frac{p_1}{w} + \frac{\sigma_1^2}{2g} = \text{a constant};$$

which is Bernouillis' equation for hydraulic flow.

In the case of a *gas*, suppose first that the expansion during flow is given by

$$pu = \text{constant}.$$

Then equation (1) becomes

$$\frac{\sigma_2^2 - \sigma_1^2}{2g} = p_1 u_1 \log_e \frac{p_1}{p_2};$$

or $$p_2 u_2 \log_e p_2 + \frac{\sigma_2^2}{2g} = p_1 u_1 \log_e p_1 + \frac{\sigma_1^2}{2g} = \text{constant};$$

that is $$\text{CT}_2 \log_e p_2 + \frac{\sigma_2^2}{2g} = \text{CT}_1 \log_e p_1 + \frac{\sigma_1^2}{2g} = \text{constant}.$$

If the expansion during the flow along the pipe follows the law

$$pu^n = \text{constant},$$

then equation (1) gives

$$\frac{n}{n-1} \cdot p_2 u_2 + \frac{\sigma_2^2}{2g} = \frac{n}{n-1} \cdot p_1 u_1 + \frac{\sigma_1^2}{2g} = \text{constant};$$

which, in the case of *adiabatic flow* when $n = \gamma$, reduces to

$$\text{K} \ . \text{T}_2 + \frac{\sigma_2^2}{2g} = \text{K}_p . \text{T}_1 + \frac{\sigma_1^2}{2g} = \text{constant}.$$

T_2 can always be calculated when the pressures p_1 and p_2 are known; for in the case of steam it will be the temperature corresponding to the pressure p_2, while for gases it will be given by

$$\text{T}_2 = \text{T}_1 \left(\frac{p_1}{p_2}\right)^{\frac{1-n}{n}}.$$

Hence in all cases the increase of kinetic energy of the gases during flow can be calculated on the assumptions made.

The weight (W lbs.) of gas passing any section (a) of the pipe per second will be given by

$$\text{W} = \frac{\sigma a}{u},$$

where u is the volume per lb. at the section (a) where the velocity is σ.

Hence if we know W, a, and the volume u_1 where the pressure is p_1, we can ascertain σ_1, and then calculate the change in velocity and so find σ_2 from the equations just given.

§ 43. **Flow of Hot Gases up Boiler Funnels.**—When the flow of hot gases up a boiler funnel is caused by "natural draught," that is to say, when it is simply due to the expansion of heated gases in the boiler furnace, the velocity of the gases is produced by the difference of pressure (Δp) at the entrance of the furnace between the weight of a column of hot funnel gases of the height of the funnel, and the weight of a similar column of atmospheric air outside it. In such a case the velocity of flow σ will be given by

$$\frac{\sigma^2}{2g} = \Delta p \times \mathrm{V},$$

where V is the mean volume per lb. of the funnel gases; for before the air enters the furnace its velocity is very small, and so the *change* of velocity may be taken as σ itself.

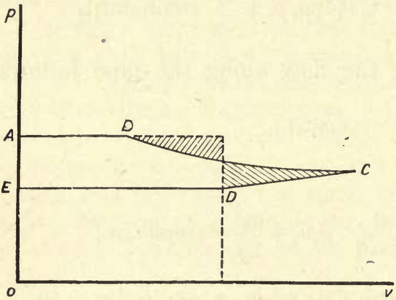

Fig. 40.

It is therefore necessary to find V and Δp for any given funnel length and temperature in order to obtain σ, and from this the weight of funnel gases passing up the funnel in a given time. If a curve be plotted to give the volumes of 1 lb. of furnace gas as it passes through the furnace, combustion chamber, tubes, and funnel, it will be as shown in fig. 40—the volume per lb. increasing and decreasing according to the temperatures of the various parts of the boiler. Thus AB will represent the volume of 1 lb. of air as supplied at the stokehold pressure and temperature; BCD the change of volume and pressure as it passes through the furnace, tubes, etc.; and DE the volume per lb. of funnel gas at the average funnel temperature.

It is found that the two shaded areas of fig. 40 are practically equal, and hence it follows that the *mean volume* V of the gases as they pass through the boiler and funnel is the volume corresponding to that of the gases when in the funnel.

Next as regards Δp. Let W be the weight of a column of funnel gases of length l (where l is the height measured from the furnace grate to the funnel top), and let W_0 be the weight of a similar column of external air at absolute temperature T_0.

Let V_0 be the volume of 1 lb. of air at atmospheric temperature T_0, and V_1 its volume at the funnel temperature T_1, and let A be the sectional area of the funnel.

Then
$$W = \frac{lA}{V}.$$

Also, if n lbs. of air be supplied per lb. of coal burnt in the furnace, it is found

$$(n+1)V = nV_1.$$

$$\therefore \quad W = \frac{lA}{\dfrac{n}{n+1} \cdot V_1} = \frac{lA}{\dfrac{n}{n+1} \cdot \dfrac{T_1}{40}};$$

for $p_1 V_1 = 53{\cdot}2 T_1$, and $p_1 = 14{\cdot}7 \times 144$ lbs. per sq. ft.

Similarly,
$$W_0 = \frac{lA}{V_0} = \frac{lA}{\dfrac{T_0}{40}}.$$

Hence,
$$\Delta p \times A = W_0 - W = 40lA\left[\frac{1}{T_0} - \frac{n+1}{nT_1}\right].$$

$$\therefore \quad \frac{\sigma_2}{2g} = 40lV\left[\frac{1}{T_0} - \frac{n+1}{n.T_1}\right].$$

Now if Ω be the weight of funnel gases, in lbs., discharged per second—

$$\Omega = \frac{\sigma.A}{V} = \frac{A}{V}\sqrt{2glV\left(\frac{1}{T_0} - \frac{n+1}{n.T_1}\right)40}$$

$$= \frac{A{\cdot}320}{\sqrt{T_1}}\sqrt{\frac{n+1}{n} \cdot l\left(\frac{1}{T_0} - \frac{n+1}{n.T_1}\right)}$$

by substitution for V.

With a given temperature T_0 we find, by differentiating the above expression and equating it to zero, that the condition for maximum draught is given by

$$\frac{T_1}{T_0} = 2\left(\frac{n+1}{n}\right).$$

Usually about 24 lbs. of air are supplied per lb. of coal burnt in a furnace, so that $n=24$.

Hence, approximately,

$$\frac{T_1}{T_0}=\frac{25}{12}.$$

Thus if the temperature of the atmospheric air be 60° F., *i.e.* $T_0=521$, then for maximum draught T_1 must be 1086, that is 625° F.

Usually funnel draught is measured by the height of water column that can be supported by the difference of pressure between the pressure of the gases in the funnel and the pressure of the external atmosphere.

A column of water 1 inch high corresponds to a pressure of 5·2 lbs. per sq. ft. Hence if i measure the difference of pressure of the atmosphere over that of a furnace, in inches of water—

$$i=\frac{\Delta p}{5\cdot 2}=7\cdot 7l\left(\frac{1}{T_0}-\frac{n+1}{nT_1}\right).$$

In practice it is found the actual draught is about 80 per cent. of the theoretical draught as thus calculated.

§ 44. **Effect of Friction on the Flow of Gases through Pipes.**—So far in dealing with flow, frictional resistances between the flowing gases and the surface of the enclosing pipes have been treated as negligible. In long pipes of small relative diameter, this assumption is by no means justified. It is found in practice that for such pipes, though the fall of pressure between the two ends of the pipe may be finite, yet the velocity of flow of the gas alters but slightly as it proceeds along the pipe. Hence the energy available, owing to the fall of pressure, must be dissipated in overcoming the frictional resistance to the flow.

It necessarily follows from the assumption that the velocity of flow (σ) is constant along the pipe, that, for a pipe of uniform sectional area, the volume per lb. of gas (u) must everywhere be the same.

Hence if Δp be the difference of pressure between the two ends of such a pipe, the head (h) available to overcome friction is given by

$$h=\Delta p \,.\, u.$$

Now, as in the case of hydraulic flow for a pipe of given internal surface s and area a, the head lost by friction may be written

$$h = f\frac{\sigma^2}{2g} \cdot \frac{s}{a},$$

where f is a *frictional coefficient.* Since $s = \pi dl$ and $a = \frac{\pi d^2}{4}$ for a pipe of length l, and diameter d,

$$\therefore \quad h = f\frac{\sigma^2}{2g} \cdot \frac{4\pi dl}{\pi d^2} = f\frac{\sigma^2}{2g} \cdot \frac{l}{m}$$

where m is equal to $\frac{\pi d^2}{4\pi d}$, and is called the "hydraulic mean depth."

Hence for small differences of pressure

$$\Delta p \, . \, u = f\frac{\sigma^2}{2g} \cdot \frac{l}{m}.$$

The value of f can be determined from this formula for any given pipe by experimenting on the weight of gas discharged for a known drop of pressure Δp along the pipe. In all cases the volume u will be the volume per lb. at the average pressure in the pipe, and will depend on the pressure and temperature in the case of such gases as air, and on the pressure and dryness fraction for such vapours as steam. Hence, knowing the pressures at the two ends of the pipe, u can be deduced, and also σ; for

$$\sigma = \frac{u\mathrm{W}}{a}$$

where W is the measured discharge in lbs. per second, l and m will be constants for any given pipe, and so f can be determined from the formula for any particular case. Various experiments made on the flow of air through the Paris mains and St Gotthard Tunnel, and on the flow of dry steam through well-lagged pipes, appear to show that the value of f simply depends on the diameter (d) of the pipe, and is very approximately represented by

$$f = \cdot 0027\left(1 + \frac{3}{10.d}\right)$$

in both cases, where d is measured in feet.

It must be remembered that the energy necessary to overcome friction reappears as eddies in the flowing stream, and these eddies will finally be absorbed and again appear as heat in the gas. Hence assuming no heat is lost by radiation or conduction in the passage of the gas through the pipe, the flow will be practically isothermal in the case of such gases as *air*; whilst in the case of *steam*, if it enter the pipe dry, it will be superheated at its exit.

As a certain amount of radiation must occur it is usual to assume that for air passing through pipes, the flow is isothermal; and for steam, that it remains dry and saturated if it starts dry on entering the pipe.

With *air* the temperature of flow is usually atmospheric, so that

$$u=\frac{53{\cdot}2\mathrm{T}}{p}=\frac{27{,}700}{p} \text{ at } 60^\circ \mathrm{F};$$

$$\therefore \quad \frac{\Delta p}{p}=\frac{\sigma^2 l}{d}\cdot\frac{f}{455{,}000} \quad . \quad . \quad . \quad . \quad (1)$$

For *dry steam*, approximately,

$$pu=253^2, \text{ or } u=\frac{64{,}000}{p};$$

$$\therefore \quad \frac{\Delta p}{p}=\frac{\sigma^2 l}{d}\cdot\frac{f}{1{,}030{,}000} \quad . \quad . \quad . \quad . \quad (2)$$

for a circular pipe of diameter d feet.

From these expressions it is evident the percentage loss of pressure along the pipe is independent of the initial pressure, and only depends on the velocity of flow.

Thus assuming steam flowing with a velocity of 100 feet a second in a pipe of length l, and diameter d, we get

$$\frac{\Delta p}{p}=\frac{lf}{d}\times\frac{1}{103};$$

or, practically, a percentage drop of pressure given by $\frac{lf}{d}$.

Taking an average value of f as ·004, this gives $\frac{{\cdot}004 l}{d}$, that is, a pressure loss of 1 per cent. due to friction for each length of pipe equivalent to 250 diameters.

It is often convenient to express the fall in pressure in a

pipe in terms of the flow in lbs. rather than in terms of the velocity. Thus by substituting

$$\sigma = \frac{uW}{a} = \frac{64{,}000\ W}{p\,.\,a}$$

in equations (1) and (2), we get for *air*, for small drops in pressure—

$$p\,.\,\Delta p = \frac{W^2 lf}{390\,.\,d^5};$$

and for *dry steam*—

$$p\,.\,\Delta p = \frac{W^2 lf}{168\,.\,d^5},$$

when p and Δp are measured in lbs. per sq. in., W in lbs. per hour, l in feet, and d in inches.

Hence for a given discharge and initial pressure the fall in pressure along a pipe can be calculated, or for a given fall in pressure the discharge can be determined.

It should be noticed that the foregoing expressions do not include any losses due to bends, etc., and only refer to steady flow.

To give the initial velocity (σ) we must either have a pump, or a sudden drop in pressure at the entrance to the pipe. The drop in pressure to start the flow can be computed by the expressions already given for the flow of gases through orifices.

CHAPTER IX.

STEAM INJECTORS AND EJECTORS.

§ 45. **Object of the Instruments.**—Steam injectors and ejectors are instruments by which a jet of steam acting on a stream of water or other fluid with which it mingles, can impart to the resultant fluid jet a velocity sufficient to overcome a pressure that may be equal to or greater than the initial pressure of the original steam-jet; the mechanical energy of the resultant jet being obtained from the heat energy of the steam.

The name *injector* is generally given when the instrument forces out the resultant jet against a high pressure, as when feeding water into a boiler, for example; and the name *ejector* when the resultant stream is delivered at atmospheric pressure.

The instruments are said to be "lifting" or "non-lifting" according as the fluid level in the reservoir from which they take their suction is below or above the instrument itself.

The construction of these instruments is simple, and will be understood from fig. 41, where S is the steam inlet and W the suction pipe. The steam, as it rushes through the nozzle *n*, produces a partial vacuum and so draws up water through the pipe W, and, mingling with it, forces it through the nozzle *m* with considerable velocity.

The instruments have few parts to get out of order, and are very convenient when only required to deal with small quantities of water. Their efficiency as pumps is, however, very low, not more than about 3 per cent. of the work that the same steam could do in a steam-engine cylinder being obtained from the steam used. Only, however, as

ejectors are they wasteful, when the heat in the steam, not used as useful work, simply goes to heat up the discharge water and is lost. In the case of *injectors* this heat goes into the boiler again with the feed, the injector, in fact, becoming a kind of feed-heater as well as pump ; so in such cases injectors may be used with considerable economy. The efficiency and proportions of the parts of an injector can only be determined theoretically by making several somewhat arbitrary assumptions.

Assuming, however, that there is no loss by friction or radiation in the injector, that the equations for the adiabatic flow of steam may be applied to the steam-jet, and that the steam-jet is at once completely condensed on contact with

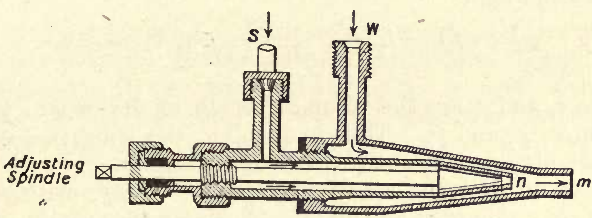

FIG. 41.—Section of a Steam Injector.

the discharge water, we can find the proper dimensions of an injector to give a given discharge against a known pressure head.

§ 46. **Calculation of Fluid Velocity through the Injector Nozzles.**—Making the assumptions stated in the last paragraph, we can find the velocity of the steam-jet at its point of contact with the incoming fluid, and the resultant velocity of the stream of water and condensed steam which leaves the outlet orifice of the injector.

The velocity of the *steam* at the point of contact will depend on the pressure and quality of the steam in the supply pipe, and on the pressure just outside the steam orifice where contact occurs between the steam and incoming water.

If p_1 and p_2 be these pressures, and q_1 and q_2 the dryness fractions of the steam at these pressures, then assuming the flow through the steam orifice of the injector to be

adiabatic, we have the velocity σ of the steam-jet given by

$$\frac{\sigma^2}{2g\,.\,\mathrm{J}}=q_1\mathrm{L}_1-q_2\mathrm{L}_2+t_1-t_2+\frac{(p_1-p_2)s}{\mathrm{J}};$$

where t_1 and t_2 are the temperatures corresponding to the pressures p_1 and p_2, and s is the volume of 1 lb. of water.

For maximum discharge through the steam orifice p_2 must be less than $\cdot 6p_1$; and it is actually found that taking $p_2 = \cdot 6p_1$ gives approximately the correct value of σ for such injectors.

Knowing p_2 we can find q_2, since the expansion in the steam nozzle is assumed to be adiabatic. It will be given by the equation

$$q_2\mathrm{L}_2=q_1\mathrm{L}_1+t_1-t_2-\frac{n}{\mathrm{J}(n-1)}(p_1q_1v_1-p_2q_2v_2),$$

where v_1 and v_2 are the volumes per lb. of dry steam at the pressures p_1 and p_2. Thus knowing q_2, the velocity σ can be at once obtained from the previous expression.

Next, to find the resultant velocity V, say, with which the *fluid* discharged by the injector finally leaves the outlet nozzle. During the contact of the steam-jet and the incoming water in the injector a loss of energy takes place simply owing to impact. Hence we cannot equate the total energy of inflow to the total energy at the outlet orifice, but must apply the principle of conservation of momentum at the point where the steam meets the water. Then we may say that:—

The momentum of the steam-jet + the momentum of the incoming water = the momentum of the resultant jet. Let y lbs. of water be pumped through by the injector per lb. of steam used in it. Then the momentum of y lbs. of water is $\frac{y}{g}\sigma_1$, where σ_1 is simply the velocity of the water at the entrance of the injector which would be produced by its head when no steam was passing through the injector. σ_1 will approximately be given by

$$\sigma_1=\sqrt{2gh},$$

where h is the head in feet of the supply water above the

injector nozzle, due either to elevation only, or to pressure, or both. The momentum of 1 lb. of steam is $\frac{1\times\sigma}{g}$. The resultant jet leaving the injector contains $(1+y)$ lbs. of mixed steam and water, and has a velocity V. Hence its momentum is

$$\frac{(1+y)}{g}V.$$

Therefore by the principle of conservation of momentum,

$$\frac{\sigma}{g}+\frac{y\sigma_1}{g}=\frac{(1+y)}{g}V;$$

$$\therefore\quad V=\frac{\sigma}{y+1}+\frac{y}{y+1}\sqrt{2gh}.$$

This gives the velocity of discharge at the outlet orifice, when the water in the supply pipe has a head h feet *above* the injector. If the supply tank be *on a level* with the injector, h may be zero. In this case the velocity of discharge becomes

$$V=\frac{\sigma}{y+1}.$$

If the supply tank be *below* the injector, the water supply must be lifted through a height h by the suction of the injector. In this case instead of adding momentum to the resultant jet, the water supply must have a corresponding momentum given to it by the steam-jet, so that in such cases the second term in the expression for V becomes negative. In any case, when the conditions under which the injector is to work are known, V can be found.

§ 47. **Determination of the Amount of Water that can be Pumped through an Injector per lb. of Steam Used.**—The number of lbs. (y) of water forced through an injector per lb. of injection steam can be determined for any given injector when the temperatures of the steam, of the feed water, and of the resultant jet are known. Let these be t_1, t_3, and t_4° F. respectively.

Then the gain of heat of y lbs. of feed water in passing from the temperature t_3° F. of the reservoir to the temperature t_4° F. of the discharge pipe is equal to $y(t_4-t_3)$ thermal units.

The heat equivalent of the kinetic energy of the jet of

water leaving the discharge orifice of the injector with velocity V is $(1+y)\dfrac{V^2}{2gJ}$ per lb. of injection steam, where $J=774$ foot-lbs.

The loss of heat of 1 lb. of injection steam at the initial temperature $t_1°$ F., condensing to the temperature $t_4°$ F., is equal to $(q_1L_1+t_1-t_4)$ thermal units.

It follows from the assumption that there is no loss of heat, and that all the steam is condensed by the time the outlet orifice of the injector is reached, that

$$y(t_4-t_3)+(1+y)\frac{V^2}{2gJ}=q_1L_1+t_1-t_4;$$

$$\therefore \quad y=\frac{q_1L_1+t_1-t_4}{t_4-t_3}$$

approximately, since the term containing V is small in comparison with the remaining terms.

The above equation will give the discharge in lbs. per lb. of injection steam when the various temperatures can be measured for any actual injector.

In *designing* an injector the resulting temperature t_4 cannot be predicted, and so y cannot be found even when the initial state of the injection steam and the temperature of the feed water are known. For design calculations it is necessary, therefore, to appeal to the equation

$$V=\frac{\sigma}{y+1}+\frac{y}{y+1}\sqrt{2gh}$$

to obtain y for any actual case instead of V. In this expression σ is known when the initial steam pressure with which the injector is to work is known. The position of the injector relative to its suction tank will also be known, and hence h will be a given quantity. V is also known when the purpose for which the injector is to be designed is decided, for the limiting value of V must be settled by the pressure head against which it is desired the injector should pump. Thus if the injector be designed to feed a boiler in which the pressure is p lbs. per sq. ft. at a height H ft. above the injector, the limiting value of V to make the feed water enter the boiler without residual velocity will be given by

$$\frac{V^2}{2g}=H+\frac{p}{w}.$$

In practice, the actual velocity of the injector discharge must be considerably greater than the velocity as thus calculated in order to overcome the weight of valves and frictional resistances in the feed pipes; but by taking a suitable coefficient the true velocity of discharge (V) can be obtained from this formula, in any given case.

Hence knowing the velocity of discharge V necessary to satisfy the given conditions, we can ascertain the number of lbs. of water (y) that can be discharged per lb. of injector steam under these conditions by substitution in the original formula.

§ 48. **The Areas of Steam and Water Nozzles necessary for a Given Discharge.**—Suppose x lbs. of water are required to be discharged per second by a given injector. Then since by supposition 1 lb. of steam will deliver y lbs. of water, the steam necessary to deliver x lbs. will be $\frac{x}{y}$ lbs. per second.

The specific volume of moist steam in the injector orifice is (q_2v_2+s) cub. ft. where the dryness fraction q_2 is determined as previously described.

Hence *the area of the steam nozzle* (a_s) of the injector is given by

$$a_s=\frac{x}{y\sigma}(q_2v_2+s) \text{ sq. ft.}$$

where σ is the velocity of the steam in feet per second.

Under these conditions the quantity of water being discharged from the injector will be $\frac{x}{y}(1+y)$ lbs. per second.

Hence the area of *the water discharge orifice* (a_w) must be given by

$$a_w=\frac{x}{y}(1+y)\frac{1}{\mathrm{V}w} \text{ sq. ft.};$$

where V is the required velocity of discharge as already calculated, and w is the weight of a cub. ft. of water.

§ 49. **Efficiency of Ejectors.**—When an injector is simply used as a pump for lifting water and discharging it at atmospheric pressure, it is known as an *ejector*.

Suppose we have an ejector at a height h_1 ft. above a tank, and it is required to draw water from the tank and deliver

it at a distance h_2 ft. above the ejector, as shown in fig. 42. Then for least loss the velocity of discharge V from the

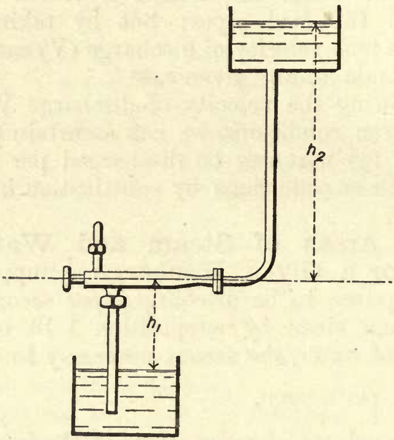

FIG. 42.

ejector must be such that no residual velocity remains as the water enters the upper reservoir. Hence

$$V = \sqrt{2gh_2}.$$

Since the suction tank is below the ejector, our equation of § 46 will then become

$$V = \frac{\sigma}{y+1} - \frac{y}{y+1}\sqrt{2gh_1};$$

or, substituting for V,

$$y = \frac{\sigma - \sqrt{2gh_2}}{\sqrt{2gh_2} + \sqrt{2gh_1}};$$

which will give y for a given steam velocity, lift, and discharge head.

It should be observed that y—that is, the water pumped up per lb. of ejector steam used—will decrease as both h_2 the discharge head, and h_1 the lift head, increase.

The efficiency of ejectors as pumps, as has already been stated, is very small, the actual work obtained from them being only of the order of ·0004 of the heat energy of the

steam used. The efficiency of lift may be increased for small lifts by using a smaller ejector than would be necessary in the ordinary way, and fitting its discharge outlet with a number of nozzles, increasing consecutively in area, as shown in fig. 43—the nozzles all being connected to the suction

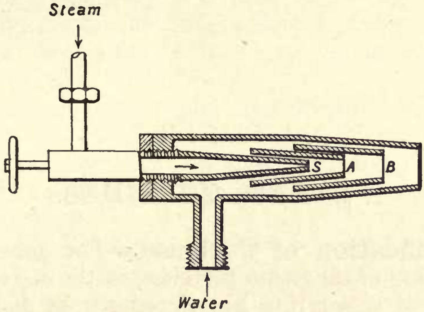

FIG. 43.

pipe. The steam nozzle *s* delivers a small stream of water at a high velocity through the nozzle A; this nozzle then acts as an ejector itself, and draws in and discharges a larger stream at less velocity through the next nozzle B, and so on. Each nozzle will thus reduce the velocity and increase the quantity of water discharged, and so finally a large amount of water may be lifted through a small height with a small ejector.

CHAPTER X.

STEAM TURBINES.

§ 50. **Classification of Turbines.**—The general principle of the working of all steam turbines, is the conversion of the heat energy of steam into kinetic energy by discharging the steam with a considerable fall in pressure, through suitable guide nozzles, on to vanes attached to one or more rotating wheels fixed to a driving shaft. The fall of pressure causes the steam-jets to issue from the guide nozzles with a high velocity, as was explained in Chapter VII., and the reaction produced by these jets striking against the vanes of the driving wheels gives the driving effort.

Broadly speaking, turbines may be divided into two classes —namely, *impulse turbines* and *reaction turbines.*

If the whole fall of steam pressure occur in the guide nozzles, and only change of *velocity* occurs during the passage of the steam through the driving wheel vanes, the turbine is said to be an "impulse" one; the driving energy depending only on the original velocity with which the steam leaves the guide nozzles.

On the other hand, if the pressure of the steam on entering the rotating wheel is reduced during the passage of the steam through the rotating vanes, the turbine is called a "reaction" turbine; for the driving effort will depend not only on the change of the original steam velocity given by the guide nozzles, but also on the change of velocity produced by the fall of pressure in the rotating wheel vanes themselves.

Each of these two classes of turbines may be further subdivided into *single-stage* turbines, in which only one set of

rotating wheel vanes is used, and into *multiple-stage* turbines. These latter may consist of a combination of a large number of successive turbines of the same type, or of a series of stages of the impulse type combined with a number of stages of the reaction type.

In the single-stage types, expansion of the steam takes place immediately to the final pressure, either in the guide nozzles alone, or else partly in the guides and partly in the vanes of the rotating wheel.

In the multiple-stage types the pressure falls in stages between each set of guides and rotating vanes, with the result that a considerably less steam velocity is produced at any one stage, and so a far less peripheral speed of rotating wheel is permissible.

A further classification of turbines is given by the direction of the flow of the steam through the vanes relative to the turbine. Thus we may have "axial" and "radial" turbines, depending on whether the steam enters and leaves the turbine wheels in a direction which is on the whole parallel to the axis of the turbine shaft, or radial to it. In the former type the velocity of the steam particles will have two components, one peripheral and the other axial, while in the latter the components will be peripheral and radial.

§ 51. **Relative Steam and Peripheral Velocities for Turbines.**—Suppose steam to be issuing from a guide nozzle with velocity σ, and let the issuing jet be directed on to a moving guide blade shaped as in fig. 44.

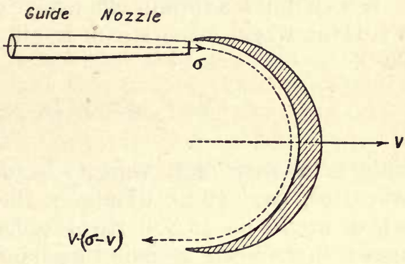

FIG. 44.

Let the velocity of this blade be V in the direction of the steam-jet. Then the relative velocity of the steam to the blade on striking it will be $(\sigma - V)$. If the blade be shaped as shown, the relative velocity with which the steam will leave the blade will be $-(\sigma - V)$, and hence the *absolute velocity* of the steam as it leaves will be $V - (\sigma - V)$.

In order that the whole of the energy of the steam-jet may be used in producing the velocity V, it is necessary that

the steam should have no absolute velocity remaining as it leaves the moving blade. Hence for maximum efficiency

$$V-(\sigma-V)=0 \text{ or } \sigma=2V.$$

That is, the velocity of the moving blade should, in this case, be half the original velocity of the steam-jet to produce the greatest effect.

Next as to the value of the velocity σ of the steam. This can be calculated for a simple orifice by the rules given in Chapter VII. For a non-conducting orifice through which the flow may be assumed adiabatic, we obtained the expression

$$\frac{\sigma^2}{2g}=\frac{n}{n-1}p_1u_1\left\{1-\left(\frac{p_1}{p_2}\right)^{\frac{1-n}{n}}\right\}$$

for the velocity of discharge from a pressure p_1 to a lower pressure p_2.

For example, if dry steam at 165 lbs. pressure per sq. in. absolute be discharged into a region at 1 lb. pressure absolute (as a condenser, for instance), the value of n will be 1·13, and then

$$\sigma=\sqrt{2g\cdot\frac{1\cdot13}{\cdot13}\cdot165\,.\,144\,.\,2\cdot72\left(1-\frac{1}{165^{\cdot115}}\right)}=4000 \text{ ft. a sec.}$$

Hence, for maximum efficiency, the peripheral velocity of a turbine wheel using steam issuing with this velocity would be

$$V=\frac{\sigma}{2}=2000 \text{ ft. per second.}$$

This is a very high velocity, and would mean that for a wheel of, say, 30-in. diameter the angular velocity of the wheel would be 15,300 revolutions per minute. This high speed introduces serious mechanical difficulties, and the whole object of modern turbine design has been to reduce this peripheral speed without loss of efficiency; though in some turbines, as the De Laval, for example, this high velocity has been adopted, the whole energy of the steam being given up in a single ring of rotating vanes.

High peripheral velocities introduce large centrifugal stresses in turbine wheels, which must be carefully designed

to withstand them. From this point of view alone, the actual peripheral speed of *solid disc* turbine wheels should not exceed 1300 ft. per second even when they are made of steel; while if the wheels are of the drum type, the peripheral speed should not exceed 350 ft. per second, unless the drums are strongly reinforced by spokes.

In actual practice the peripheral speed adopted depends not so much on the limits of strength permissible as on the duty for which the turbine has to be designed. Thus if a turbine be required to drive a dynamo, its speed of revolution will necessarily be only about 1000 to 1500 revolutions a minute in order that the dynamo may give a good efficiency. This will at once decide the diameter of the turbine wheel to give the maximum peripheral speed; and should the diameter so found be impracticable, the peripheral speed must be reduced accordingly, or else the shaft of the turbine wheel geared down as is done in the De Laval turbines. For marine work, the turbine revolutions are decided by the revolutions of the screw propeller necessary to give good propulsive efficiency. As these revolutions are usually comparatively low, peripheral speeds of only 80 to 200 feet per second can be obtained with reasonable sized turbine wheels in such cases, the speed depending on the size of ship and dimensions of screw fitted. Hence for ship propulsion it is necessary to use multiple-stage turbines, to enable the peripheral speed to be reduced without loss of economy.

With *axial turbines* it is not generally possible to direct the jets of steam in the exact direction of the motion of the wheel vanes. The discharging nozzles have therefore to be placed at a small angle with the plane of the rotating wheels, and the rotating vanes angled to allow the steam to impinge on them without shock. In practice this angle is about 17° to 20°.

The steam-jet issuing from the guide nozzles will therefore have a velocity σ in a direction making 20° (say) with the plane of rotation. Represent this velocity by the line AB in fig. 45; its component in the plane of rotation is CB. Hence, as already seen for maximum efficiency, the speed of the turbine vanes must be DB where $DB = \frac{CB}{2}$. Join AD; then AD will represent the velocity of the steam *relative* to

the rotating vanes, and the direction of AD gives the entrance slope of the wheel blades so that the steam entrance may be free from shock. The angle ADC is the angle at which the *back* of the wheel blades must be fitted on the wheel, as a shock on that side of the blades would cause greater loss than a shock on the front side. The relative velocity, represented by AD, with which the steam passes through the wheel vanes remains constant so long as friction is neglected and no change of pressure occurs in the wheel. Hence at exit the steam has the relative velocity AD, while the wheel vanes have the absolute velocity CD. Therefore if the angle of the steam discharge from the vanes be made equal to the angle of inlet, so that the velocity of discharge is given by the direction of the line DE where DE = AD, then the absolute velocity of discharge will be represented by CE. Let this be σ_1.

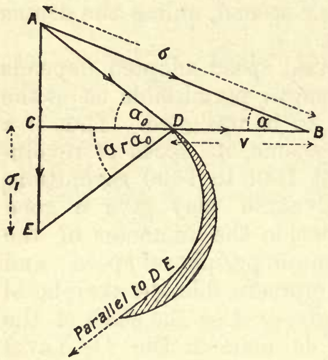

FIG. 45.

Then the total energy expended by the steam in passing through the rotating vanes is equal to $\frac{1}{2}m(\sigma^2-\sigma_1^2)$, and the total kinetic energy in the steam is $\frac{1}{2}m\sigma^2$ per unit mass. Hence the maximum efficiency of the turbine wheel is given by

$$\eta=\frac{\sigma^2-\sigma_1^2}{\sigma^2}=\frac{AB^2-CE^2}{AB^2} \text{ from the fig. 45.}$$

Hence it follows that

$$\eta_{\text{max.}}=\frac{AB^2-CE^2}{AB^2}=\frac{BC^2}{AB^2}=\cos^2 ABC=\cos^2\alpha \text{ (say)};$$

and the velocity of the wheel vanes is then given by

$$\frac{V}{\sigma}=\frac{BD}{AB}=\frac{BC}{2}\cdot\frac{1}{AB}=\frac{\cos ABC}{2}=\tfrac{1}{2}\cos\alpha.$$

Thus if V be taken at the practical limit of 1300 ft. per second, and $\alpha=17°$, then $\cos\alpha = \cdot956$.

$$\therefore \quad \eta_{\text{max.}} = \cdot 956^2 = \cdot 914\,;$$

and $$\sigma = \frac{1300 \times 2}{\cdot 956} = 2720 \text{ ft. per second.}$$

If a higher steam velocity than this be desired, then V must be taken as less than half BC, that is, less than half the resolute of σ in the direction of rotation. The corresponding value of σ_1 will then be given by

$$\sigma_1{}^2 = \sigma^2 + (2\text{V})^2 - 2\sigma\,.\,2\text{V}\,.\cos\alpha\,;$$

and the efficiency by

$$\eta = \frac{\sigma_2 - \sigma_1{}^2}{\sigma^2} = \frac{4\text{V}}{\sigma}\left[\cos\alpha - \frac{\text{V}}{\sigma}\right];$$

as will be seen by considering the triangle of velocities of fig. 46. For maximum efficiency, obviously, the lost velocity σ_1 should be as small as possible. This can be reduced by making the discharge angles of the wheel vanes as small as possible. Thus if DE in fig. 46 represent the relative velocity of the escaping steam to the wheel vanes, and the

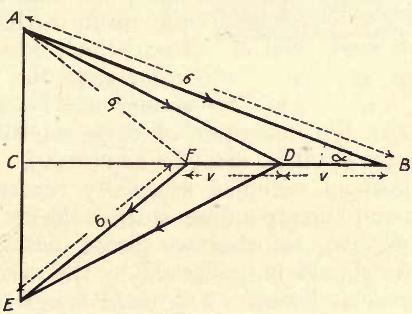

Fig. 46.

angle CDE the direction of the discharge edge of a vane, then FE represents the absolute velocity of the discharged steam, that is, σ_1; and evidently the length of FE can be reduced for a given peripheral velocity FD, and steam velocity DE, by reducing the angle CDE. In practice this angle cannot, however, be made much less than 20°, in order that the steam on leaving the rotating vanes may have a clear exit and not strike the back of succeeding vanes,

and frequently the angle CDE is made about equal to the vane inlet angle ADC. Having decided on this angle from practical considerations, it is at once seen the *least* value of σ_1 possible is CE, in which case, for maximum efficiency for a single wheel of rotating vanes, the peripheral vane speed would have to be CD, as already explained.

In some turbines, as Parsons' axial reaction turbines, a further consideration affecting the choice of peripheral vane velocity is the amount of steam leakage allowable through the clearance spaces at the tips of the vanes.

The construction adopted in the Parsons' machines is shown in fig. 47, and, in order that steam leakage losses should not be excessive, the clearance distance c should not be greater than 3 per cent. to 4 per cent. of the height h of the blades. On the other hand, the distance c must be from ·02 in. in 1 ft. diameter turbines to ·1 in. in 10 ft. turbines, for mechanical reasons, to allow for wear. Hence the minimum blade height cannot be much less than 3 per cent. of the mean turbine diameter, from which it follows that in order that the steam areas should not be too great in the first wheels, the diameters of these wheels, and hence their peripheral velocities, must be kept small.

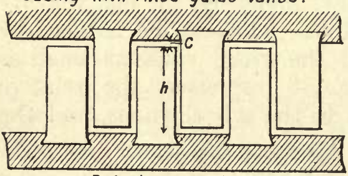

FIG. 47.

In multiple-stage turbines, especially reaction ones, the pressure drop and corresponding steam velocity through any particular wheel can be what we please, but the choice of steam velocities should be governed by the desire to obtain minimum frictional losses. Frictional losses increase with the length of the vanes and with the square of the steam velocity. Hence if σ be too large in any one wheel, the loss by friction due to it will be large, though the turbine will then only consist of a few wheels. On the other hand, if σ be small, this loss will be small; but owing to the greater number of wheels then necessary, the path of the steam will be long, and frictional losses may be considerable for this reason.

In practice a mean has to be taken, which appears to be given when the peripheral speed (V) is from ·5 to ·3 times

the steam velocity (σ) in the case of reaction turbines, and about ·36 in the case of impulse turbines.

§ 52. **The Single-stage Impulse Turbine.**—The De Laval turbine is representative of this type. It is an axial turbine, and the steam impinges on a single ring of rotating vanes from steam nozzles placed at an angle of about 20° with the plane of the wheel. These steam nozzles are designed with expanding exits, as shown in fig. 48, so that the original pressure of the steam falls to the final pressure in the nozzle itself before the steam strikes the rotating vanes. Thus no change of steam pressure occurs in the vanes of the turbine wheel, but only a change of velocity. The steam nozzles are placed symmetrically round the circumference of the rotating wheel, and any number of nozzles may be used according to the power required. Owing to the high steam velocity the peripheral velocity of the turbine wheel is also very great, and these turbine wheels run at from 30,000 to 15,000 revolutions per minute, with powers varying from 5 to 300 horse-power. In order practically to make use of this high speed, the turbine shafts are geared down in a 10-to-1 ratio by helical gearing to a low-speed shaft.

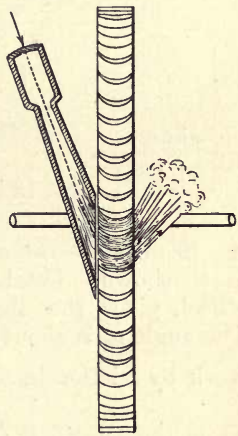

FIG. 48.—De Laval Nozzle and Wheel.

Let σ be the velocity of steam discharge through the steam nozzles when friction is neglected. Then owing to friction in the expanding portion of the nozzle, the actual velocity (σ_0) with which the steam strikes the rotating wheel is given by

$$\sigma_0 = c \cdot \sigma,$$

where c is about ·85 to ·95, σ being known when the initial and final steam pressures in the nozzles are decided.

The peripheral velocity (V) of the turbine wheel is decided from strength considerations. The relative velocity (v_0) of the steam to the wheel can then at once be found by means of fig. 49, since σ_0 makes an angle of approximately 20° with the plane of the wheel. The direction of the leading edges

of the wheel vanes is also determined by the figure. Let this direction make an angle α_1 with the wheel. The relative velocity with which the steam leaves the wheel will not be v_0 but v_1, owing to friction of the blades, where

$$v_1 = \mathrm{K} \, . \, v_0 ,$$

and K varies from ·75 to ·85.

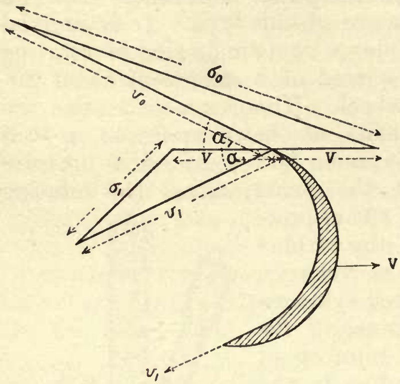

FIG. 49.

The angle with which the steam leaves the blades may have any value; but it is usual to make it the same as that of the inlet steam, but in the opposite direction. Hence v_1 in fig. 49 must be set off equal to $\cdot 75 v_0$ (say), and at an angle α_1 as shown. Combining this with the velocity V of the wheel, gives the absolute velocity of steam discharge σ_1. The angle α_1 is about 30° in De Laval turbines. The loss of work by friction in the nozzle per lb. of steam is $(1 - c^2)\frac{\sigma^2}{2g}$, while the loss in the blade channels is $(1 - \mathrm{K}^2)\frac{{v_0}^2}{2g}$.

Hence the "indicated" work per lb. of steam is

$$\mathrm{U} = \frac{c^2\sigma^2}{2g} - (1 - \mathrm{K}^2)\frac{{v_0}^2}{2g} - \frac{{\sigma_1}^2}{2g};$$

and the "indicated" horse-power will then be given by

$$\mathrm{H.P.} = \frac{\mathrm{W} \times \mathrm{U}}{550},$$

where W is the weight of steam passing through the turbine vanes per second.

The efficiency of the turbine will be given by

$$\eta = \frac{\mathrm{U} \times 2g}{\sigma^2}.$$

In designing a turbine to give a certain horse-power, U must be found as just explained, and then W can be calculated. The number of steam nozzles necessary to discharge this weight of steam per second can then be decided on by the rules for the discharge of steam through orifices. The only remaining point is the amount of divergence necessary in the

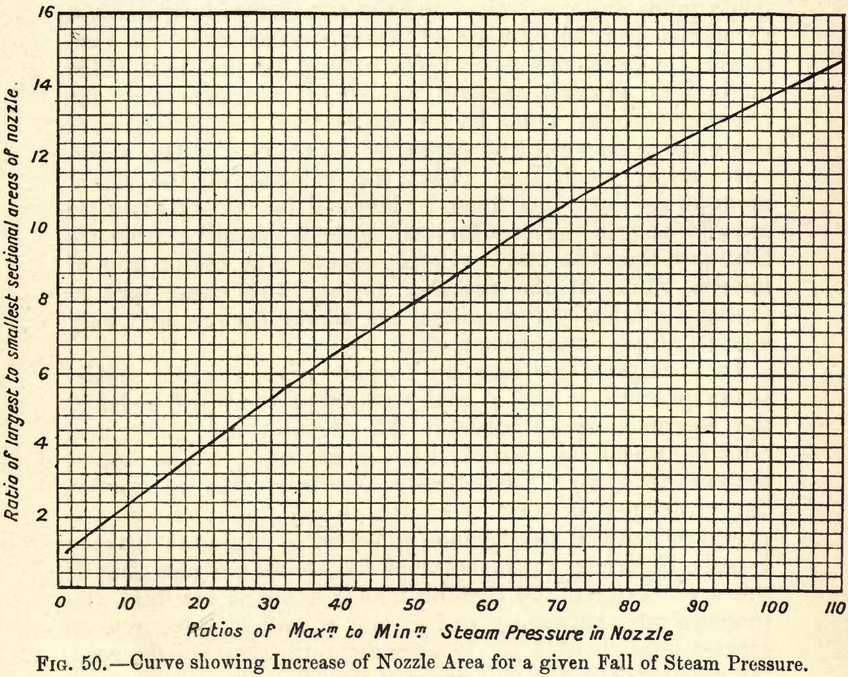

Fig. 50.—Curve showing Increase of Nozzle Area for a given Fall of Steam Pressure.

steam nozzles to give the proper degree of expansion to the steam before it strikes the rotating wheel. The ratio of the discharge area of such nozzles to the smallest sectional area of the nozzles for a given drop of pressure has been calculated by Zeuner, and is shown by the curve of fig. 50. The nozzle can be made very short up to its smallest cross-section, but after that it must diverge slowly to the final outlet area or the steam may separate from the wall of the nozzle. In practice, the angle of cone is about 10°.

§ 53. **The Single-stage Reaction Turbine.**—Reaction turbines are usually of the multiple-stage type, but the principle of their design will be more easily understood if we at first confine our attention to the reaction turbine of one stage only.

Reaction turbines usually have full peripheral steam admission—that is to say, steam is admitted through suitable guide vanes uniformly all round the rotating wheels, and not through a limited number of steam nozzles as in the case of the De Laval and other types of impulse turbines. This method is adopted because the steam entrance area for reaction turbines must be far larger than for impulse ones using the same weight of steam, as the velocity of the steam entrance is much less. That the velocity of the steam on entering the wheel must be less will be evident when we consider that in the impulse turbine the whole fall of pressure that produces the steam velocity occurs in a single ring of nozzles, while in the reaction turbine the steam falls in pressure gradually throughout the whole turbine and only a small drop occurs at each stage.

In a single-stage reaction turbine with a given initial steam pressure p and a final pressure p_1, there will exist in the clearance space between the vanes of the guide and rotating wheels an intermediate pressure p_0. The fall of pressure to p_0 may, theoretically, have any value we choose; but in practice it must be chosen to give the required peripheral velocity of the turbine wheel with minimum leakage losses. Referring to the curve of fig. 50, we see also that if the area between the turbine blades is to be constant, the pressure ratio between p and p_0, and p_0 and p_1, must not be greater than about 1·7. If a greater ratio than this be used, the area between the exit edges of the blades must be greater than between the inlet edges, which is not usually convenient. In a single-stage turbine this pressure ratio cannot in general be adopted for *both* guide and wheel vanes, as in a single stage we cannot thus reach the final given pressure; but with a number of guide and rotating wheels, as in multiple-stage turbines, this can readily be arranged. In actual practice the pressure drop per ring of vanes is far less than the above limit, and the exit areas of the blades are made *less* than the inlet areas. In Parsons' turbines it is stated that the pressure drop per stage is only about 2 lbs. per sq. in.

Taking p_0 as the pressure in the clearance space between the guide and wheel vanes, let σ_0 be the velocity of the steam at this point. Then for a given initial steam pressure p,

$$\frac{\sigma_0^2}{2g}=\frac{n}{n-1}pu\left\{1-\left(\frac{p}{p_0}\right)^{\frac{1-n}{n}}\right\},$$

since the velocity σ where the pressure is p is small, and therefore negligible compared to σ_0. The outlet angle (a) of the guide vanes and the peripheral velocity (V) of the turbine wheel are settled by the considerations given in § 51. Set off σ_0 and V at the proper angle (a), as shown in

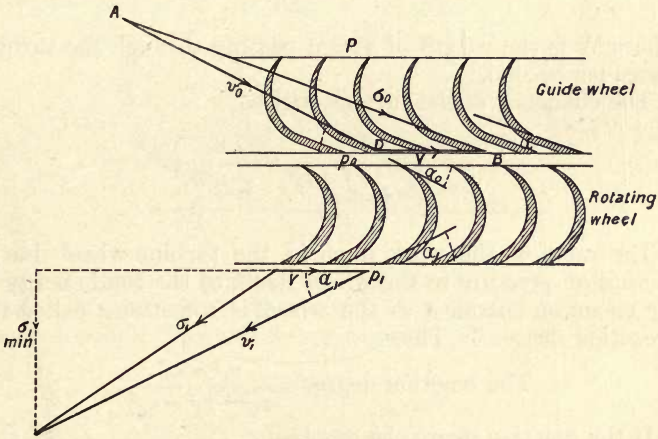

FIG. 51.

fig. 51; then the line AD gives the relative velocity (v_0) with which the steam enters the wheel vanes, and also the direction of their inlet edges—a_0 say.

In the wheel the pressure drops from p_0 to p_1, and the velocity of the steam increases to v_1 where v_1 is given by

$$\frac{v_1^2-v_0^2}{2g}=\frac{n}{n-1}p_0u_0\left\{1-\left(\frac{p_0}{p_1}\right)^{\frac{1-n}{n}}\right\},$$

where u_0 is the volume per lb. of steam at the pressure p_0.

As already seen, the exit angle of the rotating vanes is made as small as possible and generally equal to a. Setting off v_1 as thus calculated in fig. 51, and combining it with V,

we get the absolute velocity of discharge σ_1 as shown. The lost work, supposing the turbine frictionless, will then be

$$\frac{\sigma_1^2}{2g}.$$

Hence the theoretically useful work done in the turbine wheel is

$$U=\frac{\sigma_0^2}{2g}+\frac{v_1^2-v_0^2}{2g}-\frac{\sigma_1^2}{2g};$$

and the theoretical horse-power developed will be given by

$$\text{H.P.}=\frac{WU}{550},$$

where W is the weight of steam passing through the turbine vanes per second.

The efficiency of the turbine will be

$$\eta=\frac{U}{\frac{\sigma_0^2}{2g}+\frac{v_1^2-v_0^2}{2g}}=\frac{U}{U+\frac{\sigma_1^2}{2g}}.$$

The ratio of the work done in the turbine wheel due to the fall of pressure in the wheel itself, to the total energy in the steam on entrance to the wheel, is sometimes called the "reaction degree." Thus—

$$\text{The reaction degree}=\frac{v_1^2-v_0^2}{\sigma_0^2+v_1^2-v_0^2}.$$

If the reaction degree be one-half,

$$\sigma_0^2=v_1^2-v_0^2;$$

and then

$$\eta=\frac{2\sigma_0^2-\sigma_1^2}{2\sigma_0^2}.$$

With a given value of σ_0, the *minimum* value of σ_1 is when the absolute velocity of discharge is normal to the turbine wheel, when, evidently, as seen from fig. 51,

$$\begin{aligned}\sigma_{1\,\text{min.}}^2=v_1^2-V^2&=\sigma_0^2+v_0^2-V^2\\&=2\sigma_0^2-2\sigma_0 V\cos\alpha.\end{aligned}$$

Therefore the maximum efficiency is given by

$$\eta_{\text{max.}}=\frac{2\sigma_0 V\cos\alpha}{2\sigma_0^2}=\frac{V\cos\alpha}{\sigma_0}.$$

If the efficiency be unity, then $V\cos\alpha=\sigma_0$, which, when $\alpha=0$, gives

$$V=\sigma_0.$$

Hence with the *same initial steam velocity* a reaction turbine, under the above conditions, will have a peripheral vane speed twice that of an impulse turbine; for, as has already been seen for maximum efficiency with the latter type—

$$2V=\sigma_0.$$

With a known steam velocity σ_0 at the exit of the guide vanes, the area A through the vanes can be at once obtained when the power of the turbine is decided, for then the

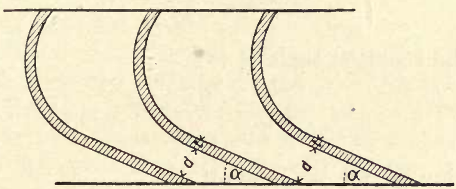

FIG. 52.

weight of steam W passing through the vanes per second is approximately known. Evidently, from fig. 52, the normal area A through the guide vanes is given by

$$A=\pi D l\sin\alpha\times\frac{d}{d+t}$$

where D is the mean diameter, and l the radial length of the blades, and d and t are as shown in the figure. Also,

$$W=\frac{A\sigma_0}{u_0}$$

where u_0 is the specific volume of 1 lb. of steam at the pressure p_0. Hence A can be obtained which gives d, when D and l are decided from considerations of the peripheral vane speed permissible.

Similarly, for the rotating wheel vanes—

$$A_1=\pi D l\sin\alpha_0\frac{d_1}{d_1+t};\text{ and } A_2=\pi D l\sin\alpha_0\times\frac{d_2}{d_2+t};$$

and
$$W=\frac{A_1 v_0}{u_0}=\frac{A_2 v_1}{u_1}$$

where u_1 is the volume per lb. of steam at the final pressure p_1 in the wheel vanes, and A_1 and A_2 are the areas at entrance and exit through the vanes.

§ 54. **Comparison of Peripheral Speeds of Single-stage Impulse and Reaction Turbines.**—Consider two single-stage turbines, one an impulse one and the other a reaction one, and suppose they develop the same power U per lb. of steam. Let σ be the initial absolute velocity with which steam enters the wheel of the impulse turbine, and σ_0 the entrance velocity to the reaction turbine. Then, as has been already seen, for the impulse turbine

$$U_{\text{max.}} = \frac{\sigma^2}{2g};$$

while for the reaction turbine

$$U_{\text{max.}} = \frac{2\sigma_0^{\,2}}{2g}$$

if the reaction degree be one-half.

Hence for the *same power*—

$$\frac{\sigma^2}{2g} = \frac{2\sigma_0^{\,2}}{2g}; \quad \text{or,} \quad \sigma = \sqrt{2}\,.\,\sigma_0.$$

From this it follows that if V be the maximum peripheral speed of the impulse turbine, and V_0 that of the reaction turbine—

$$V_0 = \frac{V\sqrt{2}}{\cos^2 \alpha};$$

for $V = \dfrac{\sigma \cos \alpha}{2}$, and $V_0 = \dfrac{\sigma_0}{\cos \alpha}$ for maximum efficiency.

In the limiting case of $\alpha = 0$, we get

$$V_0 = V\sqrt{2};$$

which means that the vanes of such a reaction turbine must run at about 1·4 times the speed of the vanes of an impulse turbine for the *same power* produced by both.

If friction be taken into account, this value will be altered; but in *any case* it must be observed that the vanes of a single-stage reaction turbine must run at a far higher speed

than the vanes of a single-stage impulse turbine for the same power.

§ 55. **Multiple-stage Impulse Turbines.**—The object of multiple-stage turbines is to obtain a low peripheral speed of turbine wheel and yet entirely use all the available energy of the steam passing through the turbine. This object may be attained in several ways with multiple-stage *impulse* turbines—viz.: (1) the steam pressure may fall to its final pressure in a single ring of guide nozzles, and the large steam velocity so produced may be gradually reduced by passing the steam through a number of successive guide and rotating wheels; or (2) only a small drop of pressure may be allowed in the first ring of nozzles and the corresponding steam velocity utilised in a single rotating wheel, the steam then being allowed to further fall in pressure in a second ring of guides and the velocity due to this second drop used in the next rotating wheel, and so on; or (3) a combination of the above two types, in which a moderate pressure drop is followed by several purely velocity stages, and then a further pressure drop is followed by several more velocity stages, until the final pressure is reached. Turbines of these types have been designed by Reidler-Stumpf, Curtis, etc.

Let us consider first the type in which a single pressure stage only is used, the peripheral velocity of the turbine being reduced by making the steam flow at constant pressure through a series of fixed and rotating vanes.

The velocities of the steam at the entrance and exit of each wheel of vanes will be given for an axial turbine by fig. 53, which shows two guide and two rotating rings. It is at once seen how the original absolute steam velocity σ_0 is reduced to a low value σ. The exit angles of the fixed guide blades is kept as small as possible—from 20° to 30°—and usually the inlet angles of the rotating vanes are made the same as their exit angles. The final absolute steam velocity σ will be given by the fact that $\sigma \cos \alpha$ must not be less than V, the peripheral vane velocity, and in practice it should be more than this. If friction be taken into account, the steam velocity through each set of vanes will decrease more quickly still, and the number of wheels necessary to bring the steam velocity to the final velocity permissible will be obtained from fig. 54. This diagram has been drawn on the assumption that the exit angle from every set of

guide blades is the same, so that the steam impinges on each rotating wheel at the same angle; and friction has been taken into account by decreasing the steam velocity by a constant proportion ($c = \cdot 8$) between each set of vanes. With the given initial steam velocity σ_0 and speed V, it is obvious from the diagram that such an impulse turbine would require three stages to reduce the steam velocity to its final amount, the number of stages in any other case depending on σ_0, on the peripheral wheel speed (V) desired, and on the frictional loss in each set of vanes.

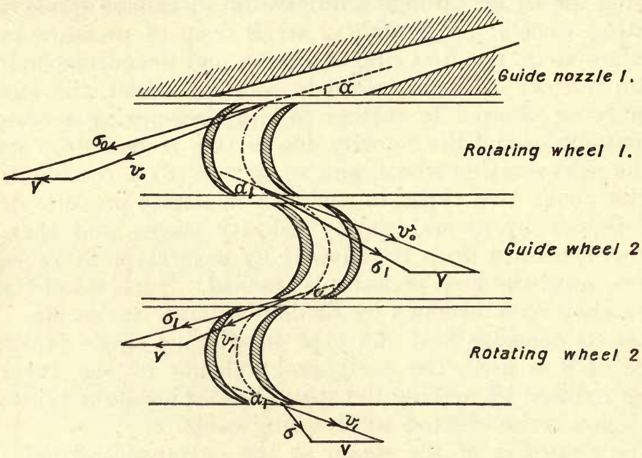

FIG. 53.

Hence in any actual case, by drawing such a velocity diagram the theoretical number of rotating and fixed wheels can be determined. It is generally convenient to draw the diagram in the more compact form shown in fig. 55. This has been drawn to the same scale as fig. 54, and by comparing the two figures it will be seen the resulting velocities and angles are identical whichever method is adopted.

Now let us consider the second type of impulse turbine, in which a single pressure stage is followed by a single velocity stage, then another pressure stage by another velocity stage, and so on. In such turbines the fall of pressure in any single ring of guide vanes will be small, and hence the velocity with which the steam enters any particular ring of

rotating vanes will also be small. The result is that a much less peripheral speed of turbine wheel can be adopted than with an impulse turbine of the single-stage type.

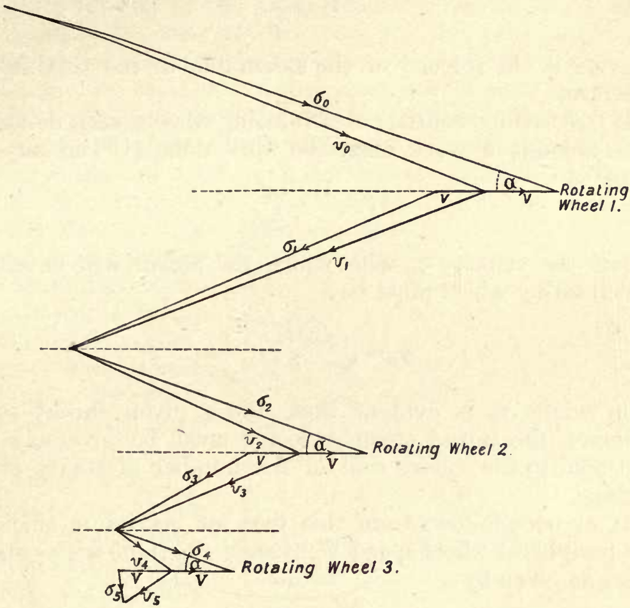

FIG. 54.

The drop in pressure in any particular set of guide vanes may be arbitrarily chosen, but from a theoretical point of

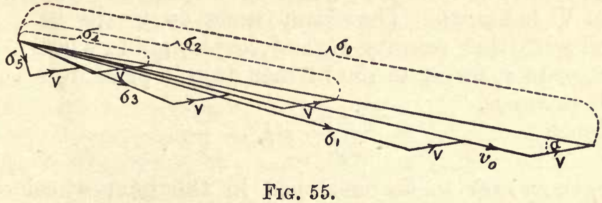

FIG. 55.

view it will be advisable to choose the pressure drops so that each wheel of rotating vanes may perform an equal amount of work.

If the whole pressure drop occur in one stage, we have already seen (§ 51) that the theoretical work available is given by

$$U = \frac{\sigma^2}{2g},$$

where σ is the velocity of the steam due to the total fall in pressure.

If our turbine consists of s rotating wheels, each doing the same amount of work, then the work done (U′) in any one wheel is

$$U' = \frac{U}{s} = \frac{\sigma^2}{2gs}.$$

Hence the velocity σ_0 with which the steam will enter the first rotating wheel must be

$$\sigma_0 = \sqrt{\frac{2gU}{s}} = \frac{\sigma}{\sqrt{s}};$$

from which it is evident that, for a given initial steam pressure, the initial steam velocity must be inversely proportional to the square root of the number of stages of the turbine.

It at once follows from this that for maximum efficiency the peripheral wheel speed V′, of such a turbine with s stages, must be given by

$$V' = \frac{V}{\sqrt{s}};$$

where V is the speed of a single-stage impulse turbine in which the initial steam velocity is σ.

The velocity (σ_1) of steam exit from the first rotating wheel can be found by the velocity diagram as seen in fig. 49, when V′ is known. The steam passes on to the next guide wheel with this velocity σ_1, and in the guide wheel σ_1 will increase to σ_2 owing to the further drop in pressure occurring in these vanes.

Hence
$$\frac{\sigma_2^2 - \sigma_1^2}{2g} = U' = \frac{U}{s},$$

since the power to be developed in the next wheel of the turbine is to be the same as that developed in the first wheel. Therefore

$$\sigma_2 = \sqrt{\sigma_1^2 + 2g\frac{U}{s}}.$$

This gives the velocity σ_2 of steam entrance to the next ring of rotating vanes; and by combining this with V′ we get the vane angles of the next ring of vanes, and the absolute velocity of the steam discharge (σ_3) from the vanes. Proceeding in this way we can get the vane angles and steam velocities for the whole *s* wheels of the turbine.

To find the cross-sectional steam areas through the various turbine wheels we must draw a theoretical expansion diagram between the given limits of pressure, as shown in fig. 56, to represent the total work U′*s* that the turbine is required to develop, and then divide it up into *s* equal areas by lines of constant pressure as shown, *s* being decided by the peripheral wheel speed desired. The horizontal division lines will then give the required drop of pressure in each ring of guide vanes, and from these pressures the specific steam volumes can be obtained that are necessary for the calculation of the cross-sectional areas of the various wheels by the method explained in § 53.

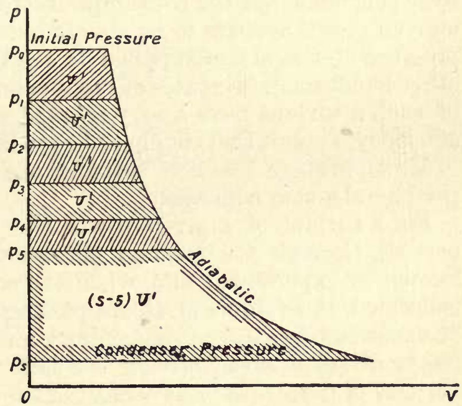

FIG. 56.

The third type of impulse turbine need not be discussed, as it simply consists of a combination of the two types just described, and therefore the same principles apply.

§ 56. **Multiple-stage Reaction Turbines.** — The principle of the action of these turbines is identical with that of the single-stage reaction type, and they may in fact simply be considered as a number of single-stage reaction turbines attached to the same shaft. The difference between the two types is, that in the single-stage turbines the total steam pressure drop between the boiler and the condenser occurs in a single pair of guide and rotating wheels, while in the multiple type the total pressure drop is divided amongst a number of wheels, with the result that the fall of pressure

in any one wheel is much reduced and the steam and peripheral velocities correspondingly decreased.

Turbines of this type have been designed by Rateau, Parsons, Schulz, and others.

The practical design of such turbines depends largely on experiment, as it is found that mechanical considerations, such as the clearance between the vanes and casings of the turbines, etc., complicate the problem in a manner that cannot be allowed for by theory alone. For instance, in axial-flow reaction turbines of the Parsons' type it is found by experiment that the consumption of steam for a turbine of a given power appears to be almost independent of the initial pressure of the steam supplied to the turbine; while on the other hand, small decreases of vacuum on the low-pressure side of such a turbine have a very marked effect in increasing its efficiency, a result that could not be predicted from theory only.

Multiple-stage reaction turbines are built only with full peripheral steam admission.

For a turbine of a given power the weight of steam (W) passing through the turbine per second is approximately known by experience; and when the condenser pressure (p_s) is decided, the volume of steam passing through the last ring of vanes can be at once determined, and from this the cross-sectional steam area through the last wheel, since the final velocity of the steam (σ_s say) must be sufficient to get the given weight of steam through the turbine—and, from practical considerations, it cannot be much less than 100 feet a second.

If p be the initial boiler pressure, the total work (U) available per lb. of steam supplied to the turbine is given by

$$U = \frac{n}{n-1} pu \left\{ 1 - \left(\frac{p}{p_s}\right)^{\frac{1-n}{n}} \right\}.$$

Hence the maximum theoretical work that can be done in the turbine per lb. of steam is

$$U - \frac{\sigma_s^2}{2g}.$$

The peripheral speed (V) of the turbine wheels must be decided on by the work that the turbine is desired to perform. When this is settled, the initial steam velocity σ_0 in the first ring of vanes is obtained from the considerations of maximum efficiency given in § 53, and from this the

fall in steam pressure in the first ring of vanes to give this velocity can be at once deduced. Assuming this *fall of pressure constant* throughout every wheel of the turbine, and knowing V, the steam velocities through any guide or rotating wheel, and the vane angles for any wheel, can be obtained graphically, identically as explained for the single-stage reaction turbine.

If the pressure drop be arranged to make the *work done in each stage constant*, then the work done in any pair of guide and rotating wheels will be the same as that done in the first pair, and will be given by

$$U_1 = \frac{\sigma_0^2 + v_1^2 - v_0^2 - \sigma_1^2}{2g},$$

where the symbols have the same meaning as before. Hence if s be the number of stages in the whole turbine,

$$s \, . \, U_1 = U - \frac{\sigma_s^2}{2g},$$

which gives s when U_1 and U are known. On either assumption the steam pressures at the commencement of every stage can be found when U, V, and σ_0 are decided; and so the specific steam volumes and cross-sectional areas through the turbine wheels can be obtained at any point, by the use of a diagram similar to that drawn for the multiple-stage impulse turbine in fig. 56.

Friction can be approximately taken into account as with the impulse turbine.

In reaction turbines of considerable power the number of stages becomes very large, and the steam area through the lower pressure wheels becomes also very great. If the peripheral speed V be kept constant for all the wheels on the same shaft, this necessitates either very large turbine blades, or else unduly large spacing of the blades in the final stages. As this is objectionable for practical reasons, it is usual in such turbines to increase the diameter of the wheels after a suitable number of stages. This increases the value of V, but permits of the required increase of steam area while keeping a reasonable length of blade.

A section through such a turbine is shown in fig. 57.

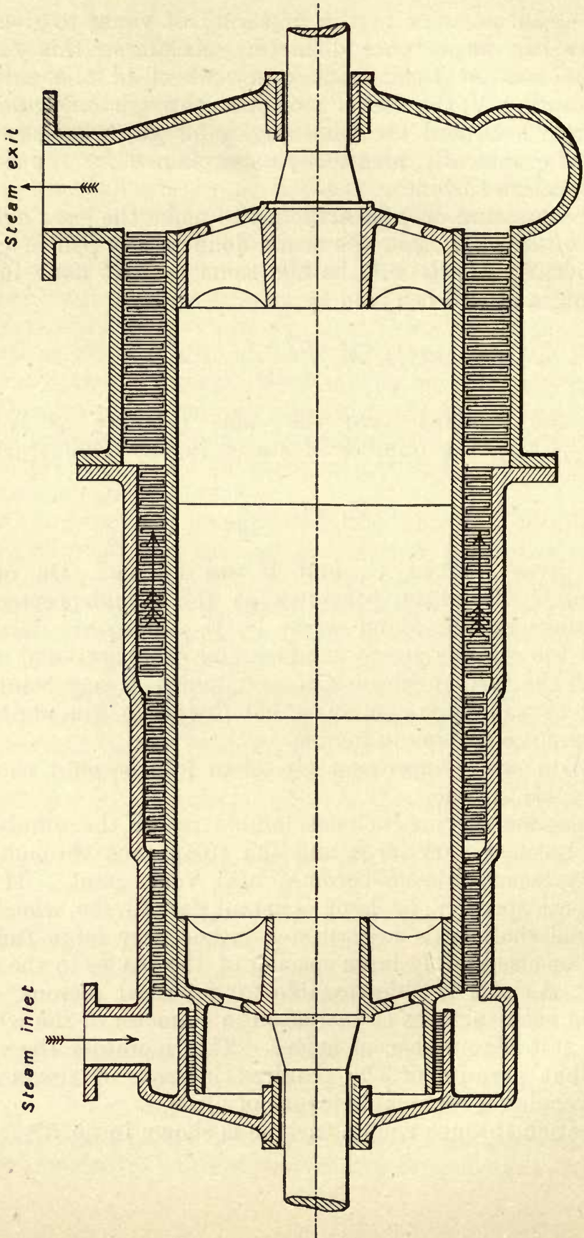

FIG. 57.—Sectional Elevation of a Parsons' Turbine.

APPENDIX I.

EXAMPLES.

CHAPTER I.

1. Find the external work done during adiabatic expansion in terms of the initial and final temperatures of the expansion.

Ans. $K_v(T_1 - T_2)$.

2. Find the store of energy in a reservoir of 5 cub. ft. capacity when air is stored in it at 70 atmospheres pressure: (*a*) when the index of the expansion law is given by $n = 1$, and (*b*) when $n = 1{\cdot}4$.

Ans. (*a*) 1080 foot-tons; (*b*) 487 foot-tons.

3. Find the absolute zero of temperature from the following data: under a pressure of 2116·4 lbs. per sq. ft. and at 32° F. the volume of 1 lb. of air is 12·387 cub. ft., while at 104° F. it is 14·2 cub. ft. *Ans.* $-461°$ F.

4. 1 lb. of air at 32° F. has its volume doubled at constant pressure. What is its final temperature, and how much external work is done during the expansion, and how much heat must be supplied? *Ans.* 525° F.; $U = 26{,}200$ foot-lbs.; $Q = 117$ B.T.U.

5. Dry steam at 15 lbs. absolute pressure per sq. in. is superheated till its temperature is 250° F., when it is assumed to be a perfect gas. In what proportion will its volume increase if the pressure remain constant? *Ans.* 1·076.

6. If 1 lb. of air does 390·6 foot-lbs. of work without receiving or rejecting any heat, what will be its fall in temperature?

Ans. 3° F.

7. The temperature of 1 lb. of air is observed to fall from 540° F. to 290° F. while it expands adiabatically to double its volume, doing 32,600 foot-lbs. of work in the process. Find the values of the specific heats of the air at constant pressure and constant volume. *Ans.* $K_p = \cdot 2375$ T.U.; $K_v = \cdot 1685$ T.U.

CHAPTER II.

8. Compare the efficiencies of: (*a*) a Stirling engine with perfect regenerator in which the maximum pressure is 115 lbs. per sq. in. absolute and minimum pressure 15 lbs. absolute, and limits of temperature 600° F. and 70° F.; and (*b*) a perfectly reversible steam-engine working between the same limits of pressure. If the piston speed and stroke be the same in both engines, compare the areas of the pistons for equal power.

Ans. 50 %; 15·7 %; $\frac{\text{Area of Stirling cylinder}}{\text{Area of steam-engine cylinder}} = 1 \cdot 17.$

9. If air expand according to the law $PV^{\cdot 592} = \text{constant}$, what fraction of the heat expended during the expansion is turned into work? *Ans.* ·5.

10. Compare and tabulate the performances of: (*a*) a perfect heat-engine, (*b*) a Stirling's air-engine, and (*c*) an Ericcson's air-engine working between limits of temperature of 650° F. and 150° F., assuming the lowest pressure is 14·7 lbs. per sq. in. and the ratio of isothermal expansion is 2 in each case, and the Stirling and Ericcson's engines are fitted with regenerators having an efficiency of ·9. Find the work done per lb. of air, the volume swept through per H.P. per minute, and the efficiency, for each engine.

	Work done.	Vol. per H.P. min.	η.
Ans. (*a*)	18,437 foot-lbs.	49 cub. ft.	·45
(*b*)	18,437 ,,	27·3 ,,	·38
(*c*)	18,437 ,,	72·5 ,,	·367

11. If a perfect air-engine work between limits of temperature of 60° F. and 500° F., find its efficiency and the ratio of the adiabatic expansion. *Ans.* ·457; 4·47.

12. Air at 60° F. is compressed from 14·7 lbs. to 1000 lbs. per sq. in. absolute pressure, according to the law $PV^{1 \cdot 204} = \text{constant}$. It is then cooled at constant volume to 60° F., and then allowed to expand according to the same law to its original pressure. Find the work done on the air per lb. in passing it through this

cycle of operations, and the number of lbs. of air dealt with per min. by a 30 H.P. pump. *Ans.* 68,100 lbs. ft.; 14·5 lbs.

13. A heat-engine receives 1000 thermal units at a temperature of 300° F., and 500 units uniformly as the temperature falls from 300° F. to 200° F. The engine rejects all its heat at 125° F. Find the maximum possible efficiency; and if the source of heat be at 500° F., what is the loss of heat due to not using the whole available energy? *Ans.* ·21; Loss = 265·2 T.U.

CHAPTER III.

14. Calculate the diagram efficiency of a gas-engine using the Otto cycle: (1) when the compression pressure is 45 lbs. by gauge, and (2) when it is 75 lbs. by gauge, per sq. in., assuming the mixture is drawn in at 15 lbs. absolute per sq. in. in each case, and the expansion and compression are adiabatic.

Ans. (1) ·345; (2) ·415.

15. Assuming 1 cub. ft. of gas requires 7 cub. ft. of air for its complete combustion, and 1 lb. of the gas occupies 30 cub. ft. at atmospheric pressure and at 59° F., find the pressure and temperature produced by the combustion of 1 lb. of the gas at constant volume, having given that the combustion produces 19,500 thermal units per lb. of gas, and that the initial temperature of the gas and air is 59° F. Assume $K_p = \cdot 248$ and $K_v = \cdot 175$ for the burnt gases. *Ans.* 195 lbs. per sq. in. abs.; 6559° F.

16. In three Otto gas-engines the compression spaces are ·6, ·4, and ·34 of the stroke volume respectively. If the actual efficiencies of the engines are ·17, ·21, and ·25, find how much of the heat expenditure is shown in the indicator diagrams, and how much is added after ignition.

	(1)	(2)	(3)	
Ans. Heat developed on ignition	·512	·525	·584	of the total
Heat added after ignition	·488	·475	·416	calorific value.

17. An Otto cycle gas-engine of 35 I.H.P. has a gas consumption of 14·7 cub. ft. per I.H.P. per hour. If the maximum pressure developed in the engine cylinder be 220 lbs. per sq. in. absolute, the compression pressure 75 lbs. absolute, and the suction pressure 14·7 lbs. absolute, find its efficiency from the indicator diagram, and also its actual efficiency, the calorific value of the gas being 19,500 T.U. per lb., and 1 lb. of the gas occupying 30 cub. ft. at atmospheric pressure. *Ans.* ·377; ·269.

18. If a cubic foot of gas having a calorific value of 19,500 T.U. per lb. be mixed with 12 cub. ft. of air at 59° F. and 14·7 lbs. pressure per sq. in. absolute, and if 1 lb. of the gas occupies 30 cub. ft. at this pressure, find the temperature and pressure produced by perfect combustion on ignition at constant volume. If the pressure actually reached be 105 lbs. absolute, find the temperature attained and the proportion of gas burnt on ignition.
Ans. 3959° F.; 125 lbs. per sq. in.; 3259° F.; 82 %.

19. What percentage of steam evaporated in a boiler is required to inject oil through burners supplying oil fuel to the boiler furnace if the pressure in the boiler be 150 lbs. per sq. in. by gauge, the temperature of the feed water be 60° F., and 1 lb. of the oil requires ·7 lbs. of steam to spray it into the furnace? Assume the calorific value of the oil to be 18,500 T.U. per lb., and that 1 lb. of the oil can evaporate 14 lbs. of boiler water from and at 212° F. *Ans.* 6·06 per cent.

20. Neglecting all losses, find the horse-power of an oil-pump to supply oil to the burners of six boilers at 300 lbs. per sq. in. by gauge — the boilers evaporating 9000 lbs. of water per hour at 300 lbs. gauge pressure from a feed water temperature of 100° F., 1 lb. of the oil being capable of evaporating 15·5 lbs. of water in the boilers from and at 212° F. *Ans.* ·24.

CHAPTER IV.

21. By means of a reversed perfect heat-engine ice at 32° F. is to be made from water at 67° F., the temperature of the brine or freezing mixture being 19° F. How many lbs. of ice at 32° F. can be made per H.P. per hour, taking the latent heat of ice as 144 thermal units? *Ans.* 143 lbs.

22. Find the least horse-power of a reversed perfect heat-engine that will make 800 lbs. of ice per hour at 27° F., from water at 70° F. Specific heat of ice = ·5. *Ans.* 5·08 H.P.

23. If the compression pressure of a reversed Joule's heat-engine be 45 lbs. absolute per sq. in., and the suction pressure be 15 lbs. absolute, find the lowest temperature produced in the engine, the air being cooled at the highest temperature by circulating water at the temperature of the atmosphere, which is 90° F. *Ans.* – 61° F.

24. In a Joule's-cycle refrigerator with a perfect interchanger, the lowest temperature produced is 367° absolute, and the

temperatures of the cooling water and air suction are both 90° F. Find the coefficient of performance, and the number of lbs. of ice that can be made per H.P. per hour. *Ans.* 2·07 ; 36 lbs.

25. A reversed perfect heat-engine is used to warm the air in a room from 32° F. to 90° F. How many cub. ft. of warm air can be supplied by the engine per H.P. per minute? *Ans.* 405.

26. An engine A working a reversed heat-engine B is employed for warming the air in a room, the actual thermal efficiency of A being $\frac{1}{8}$. B takes in heat at 32° F. and rejects it at 77° F., its relative performance compared to a perfectly reversible engine being ·6. Assuming 15 per cent. of the energy of A is lost in friction, and supposing the heat available to warm the room consists of all the heat rejected by B and $\frac{3}{4}$ that rejected by A, find the ratio of this quantity to the heat supplied to A.

Ans. 1·356.

CHAPTER V.

27. In an example of air-power transmission the maximum pressure in the compressor was 7 atmospheres, and in the air-motor 6·5 atmospheres per sq. in. absolute. Assuming the compression and expansion adiabatic, find the efficiency of the combination of compressor and motor. *Ans.* ·55.

28. In the last question if the initial air temperature be 60° F., find the final temperature in the motor: (1) supposing no drop in pressure between the pump and motor; (2) with a drop of pressure of ·5 atmospheres.

Ans. (1) $-164°$ F.; (2) $-157°$ F.

29. If V_2 cub. ft. of air have to be discharged from an air-compressor at pressure p_2 and atmospheric temperature, and if the clearance volume of the cylinder be $\frac{1}{m}$ that of the effective piston displacement, show that the volume of the cylinder must be

$$\frac{V_2}{2r}\cdot\frac{p_2}{p_1}\left\{1+\frac{1}{m}\left(\frac{p_2}{p_1}\right)^{\frac{1}{n}}-\frac{1}{m}\right\}$$

where r is the number of revolutions per minute of the compressor, and p_1 is the atmospheric pressure, the law of expansion and compression being $pV^n = \text{constant}$.

30. A pump of 1 foot stroke and pump-barrel 2 in. in diameter is used to force air into a reservoir of 1 cub. ft. volume. Find

the pressure in the reservoir after 50 strokes with the pump, and also the work done in pumping: (1) if the compression be isothermal; (2) if adiabatic.

Ans. (1) 30·7 lbs. (2) 41·6 lbs.
950 foot-lbs. 1540 foot-lbs.

31. A chamber of 100 cub. ft. capacity is exhausted by a pump to $\frac{1}{10}$ of an atmosphere. Find the least net work that must be done in pumping, assuming isothermal expansion of the air.

Ans. 142,000 foot-lbs.

32. Find the work done per lb. of air in a single-stage air-compressor in compressing air from 1 to 7 atmospheres and discharging it at the higher pressure, the initial temperature of the air being 59° F. and the law of compression $pV^{1.2}$ = constant. If the compression be carried out in two stages, what saving in work will result? *Ans.* 63,400 foot-lbs.
58,400 foot-lbs., saving 7·9 per cent.

CHAPTER VI.

33. 1 lb. of dry steam is contained in a cylinder, the pressure being 60·4 lbs. per sq. in. absolute. It is condensed at constant volume until the pressure falls to 3·5 lbs. absolute. Assuming the dryness fraction at the lower pressure is ·0686, find the heat rejected during condensation. *Ans.* 921 B.T.U.

34. In a condensing steam-engine cylinder the pressure at release is 8 lbs. per sq. in. absolute, the dryness fraction being ·8. The back pressure is 3·5 lbs. absolute, and the condenser temperature 100° F. Find the heat rejected to the condenser per lb. of steam. *Ans.* 836.5 B.T.U.

35. 1 lb. of water at 32° F. is contained in a closed vessel of 150 times its volume. Heat is supplied to the vessel until all the water is just evaporated. Find the pressure of steam produced, and the amount of heat supplied: (1) when the chamber is initially empty except for the water; (2) when the vessel is full of air above the water, and this air is expelled.

Ans. 187 lbs. per sq. in.; (1) 1110 B.T.U.; (2) 1116·5 B.T.U.

36. A boiler contains 500 gallons of water at a temperature of 60° F. Find approximately how much coal must be expended in raising steam to 60 lbs. absolute pressure, the heat given to the boiler being 10,000 B.T.U. per lb. of coal. *Ans.* 116 lbs.

37. Dry steam at 100 lbs. absolute pressure per sq. in. expands adiabatically to 40 lbs. absolute. Find the dryness fraction at the lower pressure. *Ans.* ·95.

38. In a jacketed steam-engine developing 645 I.H.P. the cylinder feed was 136·5 lbs. per minute, and the steam supplied to the jackets 6·2 lbs. per minute. The temperature of the steam supply to the cylinder and jackets was 373° F., and the feed water temperature was 90° F. Assuming all the jacket water returned to the feed tank, find the heat supplied to the engine cylinder by the jackets per H.P. per minute, and the thermal efficiency of the engine. *Ans.* 249 B.T.U.; ·17.

CHAPTER VII.

39. 10 lbs. of air at 1500 lbs. pressure per sq. in. absolute and at 59° F. escapes from a reservoir of constant volume into the atmosphere. Assuming no heat is received from the surrounding air, find the temperature of the air when it has reached atmospheric pressure. *Ans.* – 90° F.

40. Air is being discharged into a boiler furnace from an air-compressing pump at a constant pressure of 25 lbs. per sq. in. absolute. Find the energy wasted by the free expansion of the air, assuming atmospheric pressure and temperature on the low-pressure side of the air nozzles.

Ans. 16,600 foot-lbs. per lb. of air.

41. Dry steam at 100 lbs. absolute pressure per sq. in. expands adiabatically to 50 lbs. absolute, and then is released into a receiver in which the back pressure is 2 lbs. absolute. Find the dryness fractions of the steam at release and in the receiver, assuming the law of expansion is $pv^{1\cdot135} =$ constant.

Ans. ·96; ·94.

42. Dry steam at 300 lbs. absolute per sq. in. is generated in a boiler, and is then passed through a reducing valve where the pressure is reduced to 250 lbs. absolute without loss of heat. Find the dryness fraction on the lower pressure side of the reducing valve, and, if the steam be superheated, the number of degrees of superheat. *Ans.* 1·01; 14° F.

43. If steam initially dry and at 69·3 lbs. absolute pressure expand adiabatically to 14·7 lbs. absolute, deduce the value of n if the expansion law be $pV^n =$ constant. *Ans.* 1·144.

44. 1 lb. of dry steam at 100 lbs. absolute pressure is superheated by being discharged through an orifice into a region of lower pressure, the temperature of the steam after discharge being 300° F. If the steam be continuously supplied at the higher pressure, find the drop in pressure to produce this temperature of superheat. *Ans.* 68 lbs.

45. Find approximate values of $\frac{W_{max.}}{p_1 u}$ and σ_2 for steam with an initial dryness fraction of ·75 expanding hyperbolically during discharge through a simple orifice.

Ans. $\frac{1}{61}$; 1270 ft. a second.

46. A vessel containing dry steam at 315 lbs. absolute pressure discharges it into the atmosphere through an orifice 2 in. in diameter. Find the approximate weight of steam discharged per second, assuming the contracted area of the steam-jet to be ·75 times the area of the orifice. *Ans.* 10·6 lbs.

CHAPTER VIII.

47. Find the discharge in lbs. per hour of dry steam at 100 lbs. absolute pressure through a pipe of 6 in. diameter, when the pressure drop in the pipe is 1 lb. per sq. in. for each 1000 feet length of the pipe. *Ans.* 5500 lbs.

48. In a pipe 50 ft. long and 2 ft. diameter the drop in pressure is 1 lb. and the average pressure 100 lbs. per sq. in. Find the flow through the pipe in lbs. per minute.

Ans. 15,500 lbs.

49. 1000 lbs. of dry steam are to be delivered per minute through a pipe 1000 ft. long, with a drop of pressure in the pipe of 10 lbs. per sq. in. If the initial pressure of the steam be 160 lbs. absolute, find the diameter of the pipe required and the velocity of the flow. *Ans.* 8·8 in.; 114 ft. per second.

50. Dry steam at 105 lbs. absolute pressure per sq. in. expands until its pressure is 15 lbs. absolute, and is then found to be still dry. Find the heat that must have been received during the expansion per lb. of steam. *Ans.* 115 B.T.U.

51. Steam at 74 lbs. absolute pressure containing 5 per cent. of moisture is expanded hyperbolically. Find the pressure at which the steam will just become dry, and the heat received during the expansion to this pressure.

Ans. 35 lbs. per sq. in. absolute; 90·3 B.T.U. per lb.

CHAPTER IX.

52. Find the number of lbs. of water raised per lb. of steam through a total height of 100 ft. by an ejector 20 ft. above the suction tank, the steam pressure in the boiler being 45 lbs. by gauge per sq. in. and the steam dry, while the pressure in the orifice of the ejector is ·6 of the boiler pressure. Find also the work done; and if the total height of lift be halved, find the new work done and number of lbs. of water raised per lb. of steam.

Ans. (1) 11·9 lbs.; 1270 foot-lbs. per lb. steam.
(2) 16·2 lbs.; 840 foot-lbs. ,, ,,

53. If an injector receiving boiler steam at 45 lbs. gauge pressure per sq. in. be required to deliver 1200 gallons of water per hour, find the theoretical diameters of the steam and water orifices of the injector, having given that, the temperature of the feed water is 180° F., the temperature of the supply reservoir water is 100° F., and the steam pressure at the steam orifice of the injector is ·6 of the boiler pressure.

Ans. Diameter of steam orifice = ·63 ins.
,, water ,, = ·327 ins.

54. An injector feeding a boiler in which the steam pressure is 100 lbs. by gauge delivers 16 lbs. of feed water for each lb. of dry steam supplied to it by the boiler. If the initial temperature of the feed supply be 40° F., what is its final temperature on leaving the injector? *Ans.* 109° F.

55. A steam ejector is supplied with dry steam at a pressure of 231 lbs. per sq. in. by gauge. The ejector discharges water from a tank at the rate of 46·8 tons an hour through a lift of 13 ft. The temperature of the tank water is 41° F., and that of the water discharged from the ejector 83° F. Estimate approximately the quantity of steam used per ton of water pumped from the tank, and the efficiency of the ejector as a pump.

Ans. 81·75 lbs.; ·0004.

APPENDIX II.

PROPERTIES OF SATURATED STEAM.

Absolute Pressure in lbs. per square inch.	Temperature on the Fahrenheit Scale in degrees.	Sensible Heat, *i.e.* B.T.U., required to raise the Temperature of 1 lb. Water from 32° to $t°$ F.	Latent Heat, *i.e.* B.T.U., required to convert 1 lb. Water at $t°$ into Steam at $t°$.	Total Heat of Evaporation from 32° F. and at $t°$ F. in British Thermal Units = Sensible + Latent Heat.	Heat Equivalent of the External Work done during Evaporation, in B.T.U.	Volume of 1 lb. Steam in cub. ft. $v=u+$.	Weight of 1 cub. ft. of Steam in lbs.
p	t	h	L	H	$\frac{Pu}{J}$	v	w
1	102·02	70·04	1043·02	1113·06	61·62	330·4	·00303
2	126·30	94·37	1026·09	1120·46	64·11	171·9	·00582
3	141·65	109·76	1015·38	1125·14	65·65	117·3	·00852
4	153·12	121·27	1007·37	1128·64	66·77	89·5	·01117
5	162·37	130·56	1000·90	1131·46	67·66	72·56	·01378
6	170·17	138·40	995·44	1133·84	68·40	61·14	·01636
7	176·94	145·21	990·69	1135·91	69·04	52·89	·01891
8	182·95	151·25	986·48	1137·74	69·60	46·65	·02144
9	188·36	156·70	982·69	1139·39	70·11	41·77	·02394
10	193·28	161·66	979·23	1140·89	70·56	37·83	·02644
11	197·81	166·23	976·05	1142·27	70·97	34·59	·02891
12	202·01	170·46	973·10	1143·55	71·33	31·87	·03138
13	205·93	174·40	970·35	1144·75	71·66	29·56	·03383
14	209·60	178·11	967·76	1145·87	71·97	27·58	·03626
14·7	212·00	180·53	966·07	1146·60	72·18	26·37	·03793
15	213·07	181·61	965·32	1146·93	72·27	25·85	·03869
16	216·35	184·92	963·01	1147·93	72·55	24·33	·04111
17	219·45	188·06	960·82	1148·87	72·81	22·98	·04352
18	222·42	191·06	958·72	1149·78	73·06	21·78	·04592
19	225·26	193·92	956·73	1150·64	73·30	20·70	·04831
20	227·96	196·66	954·81	1151·47	73·52	19·73	·05069
21	230·57	199·29	952·98	1152·26	73·74	18·84	·05307
22	233·07	201·82	951·21	1153·03	73·94	18·04	·05545
23	235·48	204·26	949·50	1153·76	74·14	17·30	·05781
24	237·80	206·61	947·86	1154·47	74·32	16·62	·06017
25	240·05	208·89	946·27	1155·16	74·50	16·00	·06252
26	242·23	211·09	944·73	1155·82	74·68	15·42	·06487
27	244·33	213·22	943·24	1156·46	74·85	14·88	·06721
28	246·38	215·29	941·79	1157·08	75·01	14·38	·06955
29	248·36	217·31	940·38	1157·69	75·17	13·91	·07187
30	250·29	219·26	939·02	1158·28	75·32	13·48	·07420

PROPERTIES OF SATURATED STEAM—*contd.*

Absolute Pressure in lbs. per square inch. p	Temperature on the Fahrenheit Scale in degrees. t	Sensible Heat, *i.e.* B.T.U., required to raise the Temperature of 1 lb. Water from 32° to t° F. h	Latent Heat, *i.e.* B.T.U., required to convert 1 lb. Water at t° into Steam at t°. L	Total Heat of Evaporation from 32° F. and at t° F. in British Thermal Units = Sensible + Latent Heat. H	Heat Equivalent of the External Work done during Evaporation, in B.T.U. $\frac{Pu}{J}$	Volume of 1 lb. Steam in cub. ft. $v=u+s$. v	Weight of 1 cub. ft. of Steam in lbs. w
31	252·17	221·17	937·69	1158·85	75·47	13·07	·07652
32	254·00	223·02	936·39	1159·41	75·61	12·68	·07884
33	255·78	224·83	935·13	1159·95	75·74	12·32	·08115
34	257·52	226·59	933·89	1160·48	75·88	11·98	·08346
35	259·22	228·32	932·69	1161·00	76·01	11·66	·08577
36	260·88	230·00	931·51	1161·51	76·13	11·36	·08807
37	262·51	331·65	930·35	1162·00	76·26	11·07	·09036
38	264·09	233·26	929·23	1162·49	76·38	10·79	·09266
39	265·65	234·84	928·12	1162·96	76·49	10·53	·09495
40	267·17	236·39	927·04	1163·43	76·61	10·28	·09723
41	268·66	237·90	925·98	1163·88	76·72	10·05	·09951
42	270·12	239·39	924·94	1164·33	76·83	9·83	·10179
43	271·56	240·85	923·92	1164·77	76·93	9·61	·10407
44	272·97	242·28	922·92	1165·19	77·04	9·40	·10635
45	274·35	243·68	921·94	1165·62	77·14	9·21	·10862
46	275·70	245·06	920·97	1166·03	77·24	9·02	·11088
47	277·04	246·42	920·02	1166·44	77·33	8·84	·11315
48	278·35	247·75	919·08	1166·84	77·43	8·67	·11541
49	279·64	249·06	918·16	1167·23	77·52	8·50	·11767
50	280·90	250·36	917·26	1167·62	77·61	8·34	·11993
51	282·15	251·62	916·37	1167·99	77·69	8·19	·12218
52	283·38	252·87	915·49	1168·37	77·78	8·04	·12443
53	284·59	254·11	914·63	1168·74	77·87	7·89	·12668
54	285·78	255·32	913·78	1169·10	77·95	7·76	·12893
55	286·95	256·52	912·94	1169·46	78·04	7·62	·13117
56	288·11	257·69	912·12	1169·81	78·12	7·49	·13341
57	289·25	258·86	911·30	1170·16	78·20	7·37	·13565
58	290·37	260·00	910·50	1170·50	78·27	7·25	·13789
59	291·48	261·13	909·71	1170·84	78·35	7·14	·14013
60	292·57	262·25	908·93	1171·18	78·42	7·02	·14236
61	293·65	263·35	908·16	1171·51	78·49	6·92	·14459
62	294·72	264·43	907·39	1171·83	78·57	6·81	·14682
63	295·77	265·51	906·64	1172·15	78·64	6·71	·14905
64	296·81	266·57	905·90	1172·47	78·71	6·61	·15128
65	297·83	267·61	905·17	1172·78	78·78	6·52	·15350
66	298·84	268·64	904·44	1173·09	78·85	6·42	·15572
67	299·84	269·67	903·73	1173·39	78·91	6·33	·15794
68	300·83	270·67	903·02	1173·69	78·98	6·24	·16016
69	301·81	271·67	902·32	1173·99	79·04	6·16	·16237
70	302·77	272·66	901·63	1174·29	79·11	6·08	·16458

PROPERTIES OF SATURATED STEAM—*contd.*

Absolute Pressure in lbs. per square inch.	Temperature on the Fahrenheit Scale in degrees.	Sensible Heat, *i.e.* B.T.U., required to raise the Temperature of 1 lb. Water from 32 to t° F.	Latent Heat, *i.e.* B.T.U., required to convert 1 lb. Water at t° into Steam at t°.	Total Heat of Evaporation from 32° F. and at t° F. in British Thermal Units = Sensible + Latent Heat.	Heat Equivalent of the External Work done during Evaporation, in B.T.U.	Volume of 1 lb. Steam in cub. ft. $v = u + s$.	Weight of 1 cub. ft. of Steam in lbs.
p	t	h	L	H	$\frac{Pu}{J}$	v	w
71	303·73	273·63	900·95	1174·58	79·17	5·99	·16679
72	304·67	274·60	900·27	1174·87	79·23	5·92	·16900
73	305·60	275·55	899·60	1175·15	79·29	5·84	·17121
74	306·53	276·49	898·94	1175·43	79·35	5·77	·17342
75	307·44	277·43	898·28	1175·71	79·41	5·69	·17562
76	308·34	278·35	897·64	1175·99	79·47	5·62	·17783
77	309·24	279·27	896·99	1176·26	79·53	5·56	·18003
78	310·12	280·17	896·36	1176·53	79·58	5·49	·18223
79	311·00	281·07	895·73	1176·80	79·64	5·42	·18443
80	311·87	281·95	895·11	1177·06	79·69	5·36	·18663
81	312·73	282·83	894·49	1177·32	79·75	5·30	·18882
82	313·58	283·70	893·88	1177·58	79·80	5·24	·19102
83	314·42	284·56	893·28	1177·84	79·86	5·18	·19321
84	315·25	285·41	892·68	1178·09	79·91	5·12	·19540
85	316·08	286·26	892·08	1178·34	79·96	5·06	·19759
86	316·89	287·10	891·50	1178·59	80·01	5·01	·19078
87	317·71	287·93	890·91	1178·84	80·06	4·95	·20197
88	318·51	288·75	890·34	1179·09	80·11	4·90	·20416
89	319·31	289·57	889·76	1179·33	80·16	4·85	·20634
90	320·09	290·37	889·20	1179·57	80·21	4·80	·20853
91	320·88	291·18	888·63	1179·81	80·26	4·75	·21071
92	321·65	291·97	888·08	1180·05	80·31	4·70	·21289
93	322·42	292·76	887·52	1180·28	80·35	4·65	·21507
94	323·18	293·54	886·97	1180·51	80·40	4·60	·21725
95	323·94	294·31	886·43	1180·74	80·44	4·56	·21943
96	324·69	295·08	885·89	1180·97	80·49	4·51	·22160
97	325·43	295·85	885·35	1181·20	80·53	4·47	·22378
98	326·17	296·60	884·82	1181·42	80·58	4·43	·22595
99	326·90	297·35	884·30	1181·65	80·62	4·38	·22812
100	327·63	298·09	883·77	1181·87	80·67	4·34	·23029
101	328·35	298·83	883·25	1182·09	80·71	4·30	·23246
102	329·06	299·57	882·74	1182·30	80·75	4·26	·23463
103	329·77	300·29	882·23	1182·52	80·79	4·22	·23680
104	330·47	301·01	881·72	1182·73	80·84	4·19	·23897
105	331·17	301·73	881·21	1182·95	80·88	4·15	·24114
106	331·86	302·44	880·71	1183·16	80·92	4·11	·24330
107	332·55	303·15	880·21	1183·37	80·96	4·07	·24547
108	333·23	303·85	879·72	1183·57	80·99	4·04	·24763
109	333·91	304·55	879·23	1183·78	81·03	4·00	·24979
110	334·58	305·24	878·74	1183·99	81·07	3·97	·25195

PROPERTIES OF SATURATED STEAM—*contd.*

Absolute Pressure in lbs. per square inch. p	Temperature on the Fahrenheit Scale in degrees. t	Sensible Heat *i.e.* B.T.U., required to raise the Temperature of 1 lb. Water from 32° to t° F. h	Latent Heat, *i.e.* B.T.U., required to convert 1 lb. Water at t° into Steam at t°. L	Total Heat of Evaporation from 32° F. and at t° F. in British Thermal Units = Sensible + Latent Heat. H	Heat Equivalent of the External Work done during Evaporation, in B.T.U. $\frac{Pu}{J}$	Volume of 1 lb. Steam in cub. ft. $v=u+s$. v	Weight of 1 cub ft. of Steam in lbs. w
111	335·25	305·93	878·26	1184·19	81·11	3·93	·25411
112	335·91	306·61	877·78	1184·39	81·15	3·90	·25626
113	336·57	307·29	877·31	1184·59	81·18	3·87	·25842
114	337·23	307·96	876·84	1184·79	81·22	3·84	·26058
115	337·87	308·62	876·37	1184·99	81·26	3·81	·26273
116	338·52	309·28	875·91	1185·19	81·29	3·78	·26489
117	339·16	309·94	875·44	1185·38	81·33	3·75	·26704
118	339·80	310·59	874·99	1185·58	81·37	3·72	·26919
119	340·43	311·24	874·53	1185·77	81·40	3·69	·27135
120	341·06	311·89	874·08	1185·96	81·44	3·66	·27350
121	341·68	312·52	873·63	1186·15	81·47	3·63	·27565
122	342·30	313·16	873·18	1186·34	81·51	3·60	·27780
123	342·92	313·79	872·73	1186·53	81·54	3·57	·27995
124	343·53	314·42	872·29	1186·71	81·58	3·55	·28210
125	344·14	315·05	871·85	1186·90	81·61	3·52	·28424
126	344·74	315·67	871·41	1187·08	81·65	3·49	·28639
127	345·34	316·29	870·98	1187·27	81·68	3·47	·28853
128	345·94	316·90	870·55	1187·45	81·71	3·44	·29068
129	346·53	317·51	870·12	1187·63	81·74	3·42	·29282
130	347·12	318·12	869·69	1187·81	81·77	3·39	·29496
131	347·71	318·73	869·26	1187·99	81·81	3·37	·29710
132	348·29	319·33	868·84	1188·17	81·84	3·34	·29924
133	348·87	319·92	868·42	1188·34	81·87	3·32	·30138
134	349·44	320·52	868·01	1188·52	81·90	3·29	·30352
135	350·02	321·11	867·59	1188·69	81·93	3·27	·30566
136	350·58	321·69	867·18	1188·87	81·96	3·25	·30780
137	351·15	322·27	866·77	1189·04	81·99	3·23	·30993
138	351·71	322·85	866·36	1189·21	82·02	3·20	·31207
139	352·27	323·43	865·96	1189·38	82·05	3·18	·31420
140	352·83	324·00	865·55	1189·56	82·08	3·16	·31634
141	353·38	324·57	865·15	1189·72	82·11	3·14	·31847
142	353·93	325·14	864·75	1189·89	82·14	3·12	·32060
143	354·48	325·71	864·35	1190·06	82·17	3·10	·32273
144	355·02	326·27	863·96	1190·23	82·19	3·08	·32487
145	355·56	326·82	863·57	1190·39	82·22	3·06	·32700
146	356·10	327·38	863·18	1190·55	82·25	3·04	·32913
147	356·64	327·93	862·79	1190·72	82·28	3·02	·33126
148	357·17	328·48	862·40	1190·88	82·30	3·00	·33339
149	357·70	329·02	862·02	1191·04	82·33	2·98	·33552
150	358·22	329·57	861·63	1191·20	82·36	2·96	·33764

PROPERTIES OF SATURATED STEAM—*contd.*

Absolute Pressure in lbs. per square inch.	Temperature on the Fahrenheit Scale in degrees.	Sensible Heat, *i.e.* B.T.U., required to raise the Temperature of 1 lb. Water from 32° to t° F.	Latent Heat, *i.e.* B.T.U., required to convert 1 lb. Water at t° into Steam at t°.	Total Heat of Evaporation from 32° F. and at t° F. in British Thermal Units = Sensible + Latent Heat.	Heat Equivalent of the External Work done during Evaporation, in B.T.U.	Volume of 1 lb. Steam in cub. ft. $v=u+s$.	Weight of 1 cub. ft. of Steam in lbs.
p	t	h	L	H	$\frac{Pu}{J}$	v	w
151	358·62	330·12	861·23	1191·36	82·38	2·94	·33986
152	359·14	330·63	860·85	1191·51	82·41	2·93	·34199
153	359·66	331·18	860·47	1191·67	82·43	2·91	·34411
154	360·17	331·72	860·10	1191·82	82·46	2·89	·34624
155	360·68	332·24	859·72	1191·98	82·48	2·88	·34836
156	361·20	332·77	859·35	1192·13	82·51	2·86	·35049
157	361·70	333·30	858·98	1192·28	82·53	2·84	·35261
158	362·21	333·82	858·61	1192·44	82·56	2·83	·35474
159	362·70	334·34	858·24	1192·61	82·59	2·81	·35686
160	363·34	334·85	857·91	1192·76	82·62	2·79	·35889
161	363·63	335·38	857·54	1192·91	82·64	2·78	·36101
162	364·20	335·89	857·16	1193·06	82·66	2·76	·36313
163	364·69	336·40	856·80	1193·21	82·69	2·75	·36524
164	365·18	336·91	856·44	1193·36	82·71	2·73	·36736
165	365·68	337·41	856·09	1193·50	82·73	2·72	·36947
166	366·17	337·92	855·71	1193·65	82·75	2·70	·37160
167	366·65	338·41	855·36	1193·80	82·78	2·68	·37372
168	367·13	338·90	855·00	1193·95	82·80	2·67	·37583
169	367·62	339·41	854·66	1194·10	82·82	2·65	·37795
170	368·23	339·89	854·36	1194·25	82·85	2·63	·38007
171	368·59	340·38	853·95	1194·39	82·87	2·62	·38219
172	369·04	340·87	853·61	1194·53	82·89	2·60	·38430
173	369·51	341·35	853·28	1194·67	82·92	2·59	·38641
174	369·98	341·84	852·13	1194·82	82·94	2·57	·38853
175	370·45	342·33	852·60	1194·96	82·96	2·56	·39064
176	370·90	342·80	852·27	1195·10	82·98	2·54	·39274
177	371·37	343·29	851·93	1195·24	83·00	2·53	·39486
178	371·83	343·76	851·60	1195·38	83·03	2·51	·39697
179	372·29	344·23	851·27	1195·52	83·05	2·50	·39908
180	372·89	344·71	850·96	1195·67	83·07	2·49	·40120
181	373·20	345·18	850·54	1195·80	83·09	2·48	·40331
182	373·66	345·65	850·22	1195·94	83·11	2·47	·40542
183	374·11	346·12	849·89	1196·08	83·13	2·45	·40754
184	374·56	346·58	849·57	1196·22	83·15	2·44	·40965
185	375·01	347·05	849·26	1196·36	83·17	2·43	·41175
186	375·45	347·51	848·95	1196·48	83·19	2·42	·41386
187	375·90	347·97	848·63	1196·62	83·21	2·40	·41596
188	376·34	348·42	848·30	1196·76	83·23	2·39	·41807
189	376·78	348·87	847·99	1196·90	83·25	2·38	·42020
190	377·35	349·33	847·70	1197·03	83·27	2·37	·42228

PROPERTIES OF SATURATED STEAM—*contd.*

Absolute Pressure in lbs. per square inch.	Temperature on the Fahrenheit Scale in degrees.	Sensible Heat, *i.e.* B.T.U., required to raise the Temperature of 1 lb. Water from 32° to $t°$ F.	Latent Heat, *i.e.* B.T.U., required to convert 1 lb. Water at $t°$ into Steam at $t°$.	Total Heat of Evaporation from 32° F. and at $t°$ F. in British Thermal Units = Sensible + Latent Heat.	Heat Equivalent of the External Work done during Evaporation, in B.T.U.	Volume of 1 lb. Steam in cub. ft. $v=u+s$.	Weight of 1 cub. ft. of Steam in lbs.
p	t	h	L	H	$\frac{Pu}{J}$	v	w
191	377·22	349·77	847·38	1197·16	83·29	2·36	·42438
192	378·09	350·22	847·06	1197·29	83·31	2·35	·42648
193	378·52	350·67	846·75	1197·42	83·32	2·34	·42859
194	378·95	351·12	846·43	1197·55	83·34	2·33	·43069
195	379·38	351·57	846·12	1197·68	83·35	2·31	·43279
196	379·90	352·02	845·80	1197·81	83·37	2·30	·43490
197	380·23	352·45	845·50	1197·94	83·39	2·29	·43700
198	380·65	352·89	845·19	1198·07	83·41	2·28	·43910
199	381·08	353·33	844·87	1198·20	83·43	2·27	·44121
200	381·64	353·77	844·57	1198·34	83·46	2·26	·44331
201	382·1	354·1	844·3	1198·46	83·48	2·250	·4453
202	382·5	354·6	844·0	1198·59	83·50	2·236	·4475
203	382·9	355	843·7	1198·71	83·51	2·227	·4496
204	383·3	355·3	843·3	1198·83	83·53	2·217	·4516
205	383·7	355·8	843·0	1198·96	83·55	2·206	·4538
206	384·1	356·3	842·7	1199·09	83·57	2·196	·4558
207	384·5	356·8	842·4	1199·21	83·59	2·186	·4580
208	384·9	357·2	842·1	1199·33	83·61	2·176	·4600
209	385·3	357·7	841·8	1199·46	83·63	2·166	·4621
210	385·7	358·1	841·5	1199·58	83·65	2·155	·4644
211	386·1	358·6	841·2	1199·71	83·66	2·146	·4664
212	386·5	359	840·9	1199·83	83·68	2·137	·4684
213	386·9	359·4	840·6	1199·95	83·70	2·128	·4706
214	387·3	359·9	840·3	1200·08	83·72	2·119	·4726
215	387·7	360·2	840·0	1200·20	83·73	2·109	·4746
216	388·0	360·6	839·8	1200·32	83·75	2·100	·4766
217	388·4	361·1	839·5	1200·45	83·77	2·090	·4787
218	388·8	361·5	839·2	1200·57	83·78	2·081	·4808
219	389·3	361·9	838·9	1200·69	83·80	2·072	·4830
220	389·7	362·3	838·6	1200·82	83·82	2·062	·4850
221	390·1	362·7	838·3	1200·95	83·84	2·053	·4870
222	390·5	363·1	838·0	1201·07	83·85	2·045	·4891
223	390·9	363·5	837·8	1201·19	83·87	2·036	·4912
224	391·3	363·9	837·5	1201·30	83·88	2·028	·4934
225	391·6	364·3	837·2	1201·43	83·89	2·020	·4956
226	392·0	364·8	836·9	1201·55	83·91	2·011	·4977
227	392·4	365·1	836·6	1201·66	83·93	2·002	·4999
228	392·8	365·5	836·3	1201·77	83·94	1·994	·5020
229	393·2	365·9	836·0	1201·89	83·96	1·985	·5040
230	393·6	366·3	835·8	1202·00	83·98	1·976	·5062

PROPERTIES OF SATURATED STEAM—*contd.*

Absolute Pressure in lbs. per square inch.	Temperature on the Fahrenheit Scale in degrees.	Sensible Heat, *i.e.* B.T.U., required to raise the Temperature of 1 lb. Water from 32° to t° F.	Latent Heat, *i.e.* B.T.U., required to convert 1 lb. Water at t° into Steam at t°.	Total Heat of Evaporation from 32° F. and at t° F. in British Thermal Units = Sensible + Latent Heat.	Heat Equivalent of the External Work done during Evaporation, in B.T.U.	Volume of 1 lb. Steam in cub. ft. $v=u+s$.	Weight of 1 cub. ft. of Steam in lbs.
p	t	h	L	H	$\frac{Pu}{J}$	v	w
231	394·0	366·7	835·5	1202·11	83·99	1·968	·5082
232	394·3	367·1	835·3	1202·22	84·00	1·960	·5103
233	394·7	367·5	835·0	1202·33	84·01	1·952	·5124
234	395·1	367·9	834·8	1202·44	84·03	1·944	·5145
235	395·5	368·2	834·5	1202·55	84·04	1·936	·5166
236	395·9	368·6	834·3	1202·66	84·06	1·928	·5186
237	396·3	369	834·0	1202·77	84·08	1·920	·5208
238	396·6	369·4	833·7	1202·88	84·09	1·910	·5228
239	397·0	369·8	833·4	1202·99	84·10	1 904	·5249
240	397·4	370·1	833·1	1203·10	84·12	1·897	·5270
241	397·8	370·5	832·8	1203 21	84·13	1·889	·5290
242	398·1	370·9	832 6	1203·32	84·15	1·882	·5310
243	398·5	371·3	832·3	1203·43	84·17	1·875	·5330
244	398·9	371·7	832·1	1203·54	84·18	1·868	·5351
245	399·2	372·1	831·8	1203·66	84·20	1·861	·5372
246	399·6	372·4	831·5	1203·77	84·21	1·853	·5393
247	400·0	372·8	831·3	1203·88	84·22	1·847	·5414
248	400·3	373·2	831·0	1203·00	84·24	1·839	·5435
249	400·7	373·6	830·7	1204·10	84·25	1·832	·5455
250	401·1	374·0	830·5	1204·21	84·27	1·825	·5476
251	401·4	374·3	830·2	1204·32	84·29	1·818	·5497
252	401·7	374·7	830·0	1204·43	84·30	1·812	·5517
253	402·1	375·0	829·8	1204·54	84·31	1·805	·5538
254	402·4	375·3	829·5	1204·65	84·32	1·798	·5559
255	402·7	375·7	829·2	1204·75	84·33	1·792	·5580
256	403·1	376·0	828·9	1204·86	84·34	1·785	·5601
257	403·5	376·3	828·6	1204·97	84·35	1·778	·5622
258	403·9	376·7	828·3	1205·08	84·36	1·772	·5643
259	404·2	377·1	828·0	1205·19	84·38	1·765	·5664
260	404·6	377·4	827·8	1205·29	84·39	1·759	·5685
261	404·9	377·8	827·6	1205·39	84·40	1·752	·5705
262	405·2	378·2	827·3	1205·50	84·42	1·747	·5725
263	405·5	378·5	827·1	1205·60	84·43	1·740	·5746
264	405·8	378·8	826·8	1205·70	84·44	1·733	·5767
265	406·1	379·2	826·6	1205·80	84·45	1·728	·5788
266	406·5	379·6	826·3	1205·90	84·47	1·721	·5810
267	406·9	379·9	826·0	1206·00	84·48	1·716	·5830
268	407·2	380·2	825·7	1206·11	84·49	1·710	·5851
269	407·5	380·6	825·5	1206·21	84·50	1·703	·5872
270	407·8	381·0	825·3	1206·31	84·51	1·697	·5894

PROPERTIES OF SATURATED STEAM—*contd.*

Absolute Pressure in lbs. per square inch.	Temperature on the Fahrenheit Scale in degrees.	Sensible Heat, *i.e.* B.T.U., required to raise the Temperature of 1 lb. Water from 32° to t° F.	Latent Heat, *i.e.* B.T.U., required to convert 1 lb. Water at t° into Steam at t°.	Total Heat of Evaporation from 32° F. and at t° F. in British Thermal Units = Sensible + Latent Heat.	Heat Equivalent of the External Work done during Evaporation, in B.T.U.	Volume of 1 lb. Steam in cub. ft. $v=u+s$.	Weight of 1 cub. ft. of Steam in lbs.
p	t	h	L	H	$\frac{Pu}{J}$	v	w
271	408·1	381·3	825·0	1206·41	84·52	1·691	·5915
272	408·4	381·7	824·8	1206·51	84·54	1·685	·5935
273	408·8	382·0	824·6	1206·61	84·55	1·680	·5956
274	409·1	382·3	824·3	1206·71	84·56	1·674	·5978
275	409·4	382·6	824·0	1206·81	84·58	1·668	·5999
276	409·8	383·0	823·8	1206·91	84·59	1·663	·6020
277	410·0	383·3	823·6	1207·02	84·60	1·657	·6040
278	410·4	383·6	823·4	1207·12	84·61	1·650	·6060
279	410·8	384·0	823·1	1207·22	84·62	1·645	·6080
280	411·1	384·3	822·9	1207·32	84·63	1·639	·6101
281	411·4	384·7	822·7	1207·42	84·64	1·633	·6122
282	411·8	385·0	822·5	1207·51	84·66	1·628	·6143
283	412·1	385·3	822·2	1207·61	84·67	1·623	·6164
284	412·4	385·6	822·0	1207·70	84·68	1·617	·6185
285	412·7	386·0	821·7	1207·80	84·69	1·612	6206
286	413·0	386·3	821·5	1207·90	84·70	1·606	·6227
287	413·4	386·6	821·3	1208·00	84·71	1·601	·6248
288	413·7	387·0	821·0	1208·10	84·72	1·596	·6269
289	414·0	387·3	820·8	1208·20	84·73	1·590	·6290
290	414·3	387·6	820·6	1208·30	84·74	1·585	·6310
291	414·6	387·9	820·3	1208·40	84·75	1·580	·6330
292	415·0	388·2	820·1	1208·49	84·76	1·575	·6351
293	415·3	388·5	819·9	1208·59	84·77	1·570	·6371
294	415·6	388·8	819·7	1208·68	84·78	1·565	·6392
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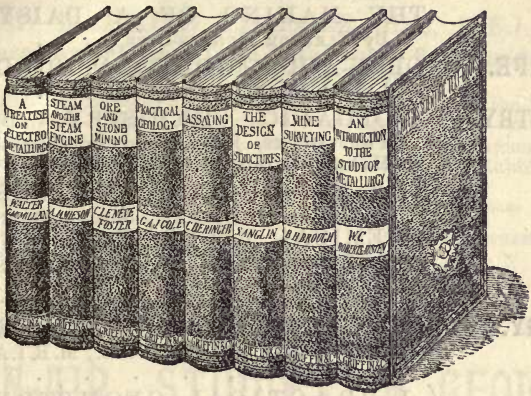

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